Heating and Ventilating

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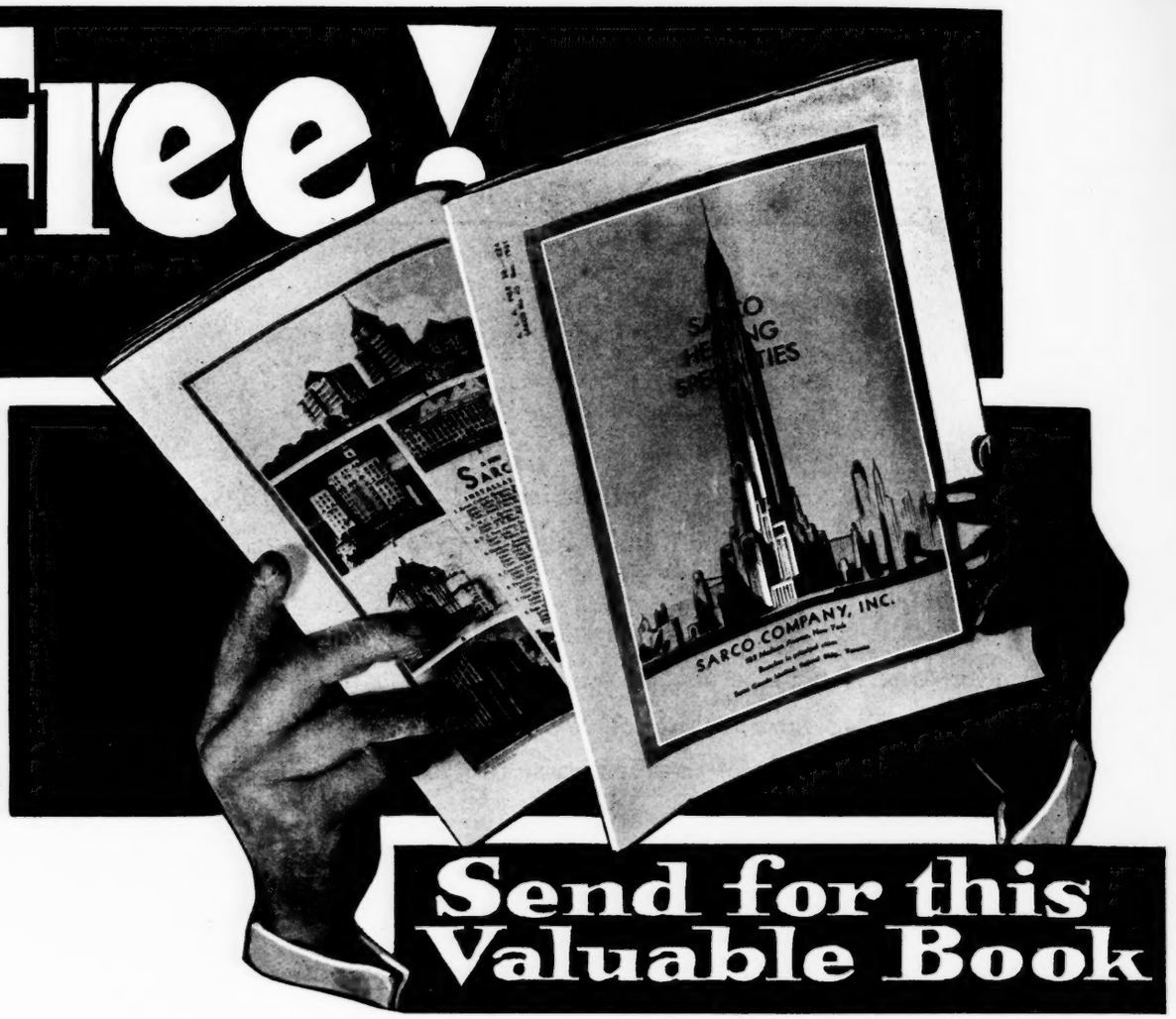

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EDITORIALS

Wage Scales and Building

PROPOSALS have recently been made in a number of cities that the union scale of wages in the building trades be reduced. If rumors are true, the union scale has been maintained in but few places, and labor and employers have found ways of avoiding the scale. The whole situation is complex. In Chicago the unions voluntarily accepted a cut of 20% in the published scale. The few days following the publication of this news saw remarkable increases in the building permits issued. No doubt some of these permits were being held back while the wage scale arrangements were pending. On the other hand, the lower scale actually may have stimulated building to some extent. We hear much of the effect of psychology, and if prospective builders know of wage scale reductions they may be more disposed to go ahead with their projects. If this proves to be true, the acceptance of wage scale reductions by labor may be well worth while. It is doubtful if labor really receives less after the scale reductions than before, and if a notice of reductions will actually stimulate building the unions can well afford to actively promote such reductions instead of opposing them.

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Heating Calculations of the Future

A RECENT speaker predicted in the course of more extended remarks that "in ten years' time we will have completely abandoned the present methods of calculation of heating."

The present methods are themselves largely the result of the calculator wishing to make sure that he was amply safe in his estimates. This object they seem to accomplish. In fact it is not at all uncommon for an installation to work satisfactorily with quantities of heating surface from 20% to 30% below the calculated amount. With competitive bidding, and where there are no specifications other than a guarantee to maintain 70°, a favorite place of cutting quantities has been to reduce the installed surface below that calculated as necessary. The fact that it is possible to do this is in itself evidence that the present accepted methods are neither correct nor logical.

The methods of calculating now in use are without doubt greatly to be preferred over the almost totally trial-and-error, or experience, methods which represented the best practice of thirty years ago. Compared with them they are complex and laborious, but they do put the calculator well on the safe side, and they are safeguards against failure due to inadequate quantities. Thirty years ago when the newer methods were first gaining a foothold the highbrows who advocated their use were regarded as visionary. Those were the days when the B.t.u. was looked on with suspicion and mystery. One has only to note its glib use now to realize how far we are from those days and those practices.

Looked at in this light it will not be so revolutionary if a new method of calculation gradually comes into use just as the present method gradually gained ground. Enough short cuts and rules have grown up around the present method so that now it is not mentally fatiguing. Probably the same will be true of a newer method before it will be generally adopted.

The first striking fact about the newer methods likely to come is that while the older methods often resulted in inadequate estimates, and the present method represents a swing to the opposite extreme with extra large estimates, the newer ones are likely to represent middle ground between them. Having swung both ways, the pendulum will stop at the middle.

Already we have evidence that newer methods are in the formative stage. One has been proposed and explained in some detail. It makes the people and objects in the heated room the basis of the calculation. It has not been proved out as yet and if it survives and develops it will be only by virtue of added refinement, clarification and data.

A second method is still more nebulous just now but, nevertheless, gradually being recognized. It will take recognition of the fact that the daily heating and cooling load is cyclic in nature. The temperature ordinarily rises during the daylight hours and falls at night. As this takes place heat flows in and out of building walls and their temperatures rise and fall. To use the method, it will be necessary to know the quantities actually leaving the heated space during the worst condition. This quantity will then become the basis of the calculation. This method offers much promise. With reasonable growth it seems that ten years might easily see this method reach a point where it can be used. The great obstacle is the plain inertia which so often retards such movements.

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And What About the Weather?

TO THOSE who are frantically seeking rising curves, plus signs in production, and per cent gains over last year, we submit some facts about the weather for the past five months.

This winter one scarcely needs charts to learn that the temperatures are abnormally high. The budding

leaves, green grass, and low fuel bills tell us that the winter is most unusual, particularly as regards record-breaking high temperatures.

Not that there are no other features which will make this winter's weather long remembered. For instance, in New York, during January, the average barometer reading was higher in the evening than in the morning —a most unusual condition, for usually the reverse is true. Too, the relative humidity at mid-day averaged 8% higher than normal.

Then there is the Southern California situation. Here is a poser for the keenest mind. To the weather man, the season is unusual, but the California booster must turn to another word—or perhaps must coin one—to express his opinion of a winter which brings to that usually mild climate a winter of frost and snow.

It is fascinating to discover that in our largest city the mean monthly temperature for January was the highest on record for 61 years; that the maximum monthly temperature surpassed all previous records; that there were only 7 days below freezing, whereas there are normally 24; that three consecutive days were 30°, 29° and 27° above normal.

To the heating engineer, however, it is more to the point to say that in the cities of Boston, New York, Pittsburgh, St. Louis and Chicago, the number of degree days from September 1 to January 31 was from 25% to 30% below normal, while San Francisco is about normal. Since the fuel consumption is roughly proportional to these percentages, it appears that the fuel interests are experiencing a disastrous season.

A wise department head with these interests will have accurate temperatures or degree day records to justify his loss of volume to his superiors. The building operator of a new building will do likewise so as to be prepared for the coming normal seasons when he will be asked why his fuel costs are so high as compared with the season of 1931-2.

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Developments in Domestic Air Conditioning

EACH of the two expositions of heating and ventilating which have been held has offered a place where one might note the course of events in the development of apparatus. At the Philadelphia show the large number of oil- and gas-burning devices was the outstanding feature and showed the interest and attention being given to the development of the use of oil and gas. At the recent Cleveland show apparatus for applying air conditioning domestically was conspicuous and showed the increasing interest in this field.

Use of air conditioning in residential heating may be said to be passing through the stage where every one is convinced that there is a great market to be had, and inventors and designers are intent upon perfecting apparatus to such a point that it can enter the field. Probably some of the attempts are rather frantic. It would be strange if this were not true.

No one can conclude now how far the potential market can be sold at present with the existing methods and equipment. Neither can anyone predict with any certainty what the line of development will be, or what standards of acceptance the public will set up. Attempts to make such estimates are rather idle. That definite, concrete, and acceptable methods and equipment will eventually come into existence is a generalization which has behind it the weight of experience in all such movements. In the meantime the appearance of so many pieces of equipment is a healthy sign of growth.

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Three Views on Ventilation Legislation

ATTEMPTS to reduce ventilation requirements to writing or to state desirable standards in language are still as difficult of accomplishment as ever, if indeed not more so. The reason is not hard to find. Ventilation is still too empirical, and too little is definitely known about it to make it easy to set down the desired characteristics in specific terms.

Three recent events have again called attention to how great a difference of opinion there still is on this subject. Several months ago the New York Commission on Ventilation set forth its ideas as to what would constitute a desirable wording for requirements for school buildings. In January the proposed building code for New York appeared and set up a statement and rules for computing for all classes of buildings. Still more recently the committee on ventilation standards of the A.S.H.V.E. presented a report embodying its ideas. It is interesting to note that none of these proposals has passed the preliminary stages.

In the case of the New York Commission on Ventilation report, it seems evident that it is to be made the basis of an attempt to amend the existing State law. The proposals must, therefore, still pass through the gauntlet of criticism almost certain to arise when public hearings are held.

The proposed building code has merely been accepted and must still be enacted before it bears the stamp of public authority.

The report of the ventilation standards committee was "received" by the society and remains to be acted on.

Here, then, we have the unpleasant situation of three separate pronouncements, all more or less aiming at the same general object of trying to reduce ventilation requirements to writing. To see how differently these groups look at the subject it is only necessary to read the three statements.

The general desirability of having a single statement which could be accepted by all is too evident to need comment. The difficulty of securing it is constantly becoming more evident. The simple truth is that existing facts do not permit agreement. Until such time as they do, legislation on the subject must be so drawn as not to block or retard the use of new knowledge as fast as it develops.

Vacuum Heating Pumps—Their Installation and Operation

By JOHN A. MASEK

A brief explanation of the function and rating of a vacuum pump, with some detailed do's and don'ts regarding the layout of the pump and its connected piping. The author also makes some valuable suggestions concerning the foundation for the pump and the use of instruments in the return line.

VACUUM pumps for steam heating have been in use for some years. While there has been a tendency on the part of some to avoid their use, recent developments both of the pumps themselves and of methods of steam heating lead to the thought that their use is likely to increase in the future rather than decrease. Of late years, the possibility of using the pumping unit as a control device has come to be better appreciated by many operating and designing engineers.

Function of the Pump

The chief function of the pump when connected to the return line of a heating system is to remove gases and condensate by mechanical means. Their positive removal induces circulation and tends toward prompter functioning of the system.

The material removed by the pump consists of gases, mainly air, and liquid water. The pump, having removed these substances from the system must dispose of them and, moreover, must do both of these acts continuously.

The removed gases are unwanted in the system and are usually expelled to the atmosphere. The liquid water is usually worth retaining in the system and is sent along its way to the boiler to be used over again to form steam.

Every pumping unit, therefore, must be able to:

1. Produce a low pressure or vacuum;
2. Receive the mixed air and water coming to it under these pressures and separate them;
3. Compress the rarified air to a pressure at least slightly above atmospheric and discharge it to the open air;
4. Pass the water from the low pressure region into one at a pressure above that of disposal. In case the water is sent directly to the boiler, this pressure must be greater than boiler pressure.

It is evident that every pumping unit is thus a vacuum producer, a water pump, and an air compressor all combined. Vacuum heating pumps differ from each other in the devices and methods used in accomplishing these objects. A great many variations are possible in methods, construction, and operating principles used. As a result, there are many types of pumps available on the market but they are not described here.

Most vacuum heating pumps now use electricity as a source of power, and are also alike in that they use a rotating element as a compressing device. They use a variety of principles to produce a vacuum.

Rating of Vacuum Pumps

Vacuum heating pumps are now rated on a basis of equivalent square feet of direct radiation which they can serve, on the assumption of an emission of 240 B.t.u., condensing 0.25 lb. of steam, per hour. The air capacity is computed on an average of 0.03 cu. ft. per sq. ft. of radiator, or 0.3 cu. ft. for every 100 ft. The air capacity is a variable factor with most heating systems, and it seems best to depend on an equivalent square feet of direct radiator rating, for the present at least. Most manufacturers state their ratings in these terms.

In determining the pump size, the total amount of direct radiator surface, or its equivalent, must be known. Since 1 sq. ft. of direct surface is based on an emission of 240 B.t.u., or 0.25 lb. of condensation, per hour, the equivalents can be calculated, where the total B.t.u. or condensate rate is given for the various units or heaters. Where heating-up periods may cause a flooded condition in the returns, it is desirable to use a factor of 0.33 lb. of condensation per sq. ft. of radiator.

Fan blast heaters, such as vento, with forced air circulation, will have a condensate rate of two to eight times that of direct radiators per square foot. Hot water storage heaters will condense an average of 1 lb. of steam per gallon of water per hour, or the equivalent of 4 sq. ft. of direct radiator, for each gallon of storage capacity, based on an average rise of 150°. The exact ratio should be determined in each case on a B.t.u. basis.

Sudden peak periods must be considered. In the case of instantaneous heaters, the condensate rate may

be twice the normal capacity at times, due to momentary demands. The air discharge from water heaters is not great.

Pressure Requirements

The boiler feed water discharge pressure must be great enough to overcome the water head pressure between the water line of the boiler and discharge outlet of the pump, plus the friction in the piping, plus the probable maximum steam pressure that may be carried at the boiler. Where the water level of the boiler is above the pump discharge, about 1 lb. of pressure is required to raise the water every 2 ft. in height.

It is recommended that the discharge piping to the boiler be as direct as possible to cut down the friction. Table 1 has been compiled to illustrate how much added

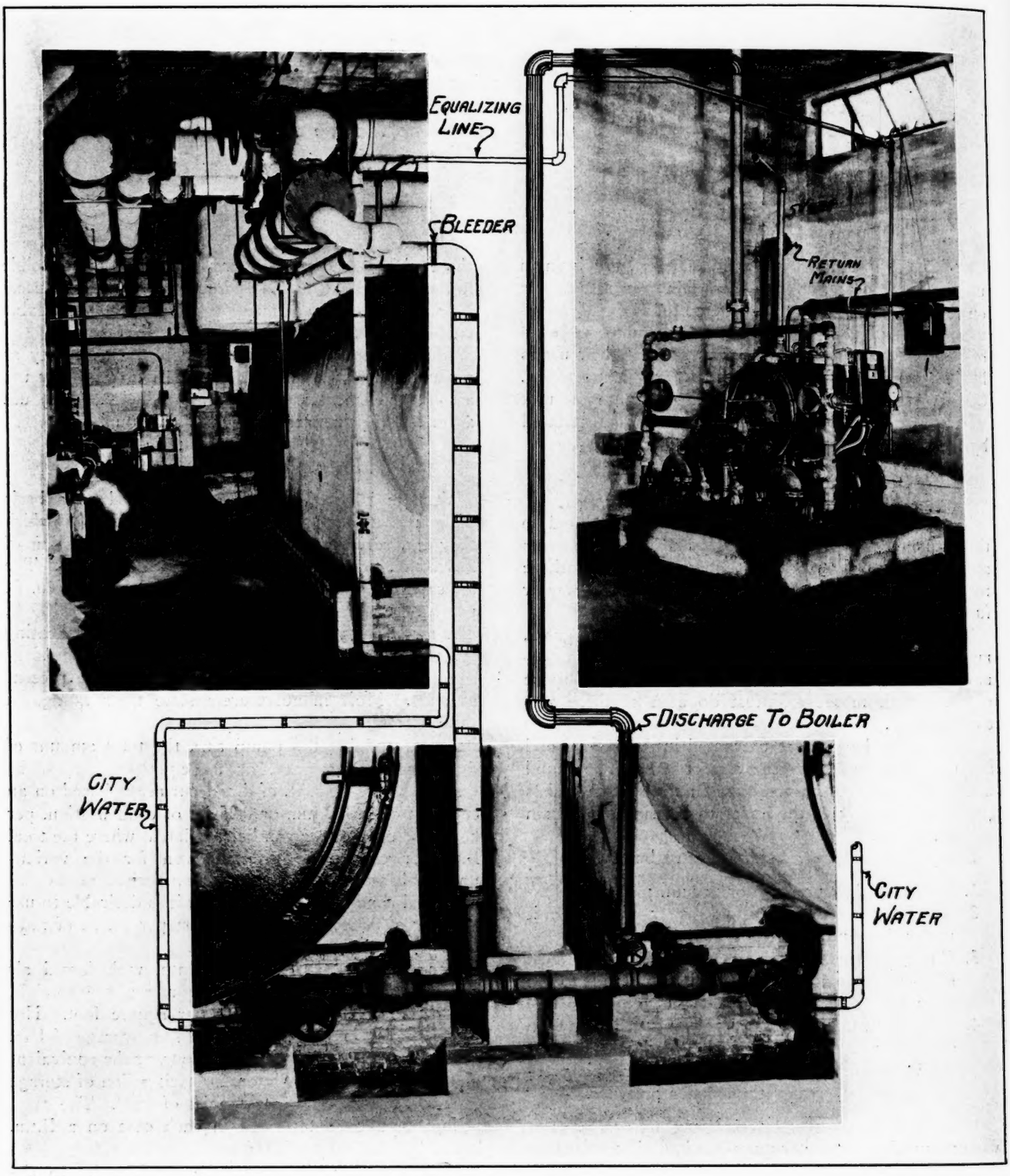


Fig. 1. A duplex jet-producing type vacuum heating pump . Note how the pipe connections are made at the boiler.

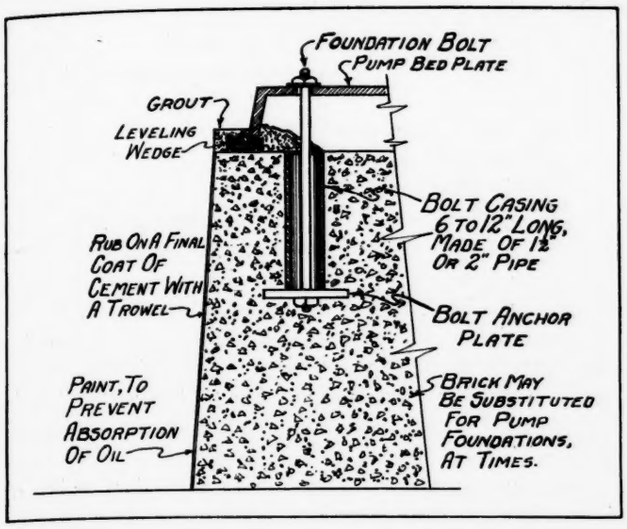

FOUNDATION BOLT
PUMP BED PLATE
GROUT
LEVELING WEDGE
BOLT CASING 6 TO 12" LONG, MADE OF 1½ OR 2" PIPE
RUB ON A FINAL COAT OF CEMENT WITH A TROWEL
BOLT ANCHOR PLATE
PAINT, TO PREVENT ABSORPTION OF OIL
BRICK MAY BE SUBSTITUTED FOR PUMP FOUNDATIONS, AT TIMES.

Fig. 2. The author's suggestion for a pump foundation

pump pressure is needed to overcome pipe friction under certain discharge conditions, and especially to illustrate the change of friction with change of pipe size.

TABLE 1.

Pipe Size, in.	1	1¼	1¼	1½	2	2	2½	2½	3	4
Flow of water in gal. per min.	10	15	25	40	40	60	100	150	150	150
Friction loss in lb. per 50 ft. in length of pipe..	1.58	1.19	3.2	3.26	0.8	1.75	1.6	3.5	1.42	0.35
Friction loss in lb. equivalent to friction through 10 elbows.....	1.05	1.00	2.7	3.26	0.1	2.31	2.7	5.8	2.85	0.92
Total Friction, lb.	2.63	2.19	5.9	6.52	0.9	4.06	4.3	9.31	4.27	1.27

Examination of the columns of the table in which the amount of water being handled is the same while the pipe size varies brings this point out well. Therefore, not only should the line be made as direct as possible but it should be of a large pipe size if friction is to be kept down. Such a consideration leads to the recommendation sometimes made that if the discharge pipe is over 20 ft. long the discharge piping should be installed one pipe size larger than the pump outlet. A reducer located close to the pump is used to make the increase in pipe size.

Many heating systems are designed to carry a boiler pressure from atmospheric up to 2 lb. to 5 lb. In such cases it is seldom that the pump is so set relative to the boiler that there will be less than 3 lb. to 4 lb. of pressure at the pump due to the difference in elevation between the pump discharge and the water line of the boiler. Under these conditions one might reasonably expect that pumps would have to work against a pressure of about 8 lb. or 9 lb. Frequently, however, the one in charge of the boiler has other duties to perform which will take him away from the boiler room. In that case he is likely to put a heavy fire in the boiler and leave it. Unless the pressure control mechanism is sensitive, the steam pressure may rise to 5 lb., 6 lb., or 7 lb., or even more. For this reason, and to allow a safety factor, a pump capable of operating against 20 lb. pressure is often advisable instead of a 10 lb. pump for the average low pressure heating job.

When to Use a Duplex Pump

It is good practice to use duplicate equipment. While one unit may suffice, a breakdown will cause inconvenience, and when the vacuum pump is a boiler feeder, it will be necessary to watch the boiler water line, as the water must be fed by hand. The installation of an automatic water feed regulator in a case of this kind would be an advantage (See Figs. 1 and 4), as the condensate is wasted into the sewer.

Since most duplex units are compact through the use of one tank or receiver, the extra space occupied will be of little consequence.

Location of Vacuum Pumps

While the proper locating of vacuum heating pumps may seem elementary, an inspection of installations will often reveal the fact that little or no thought was given to it. A frequent location is in some dark corner back of a boiler with congested piping, and where the operator must squeeze in between the boilers to get to the pump. Vacuum heating pumps of today require little attention or maintenance, but this fact does not justify sticking them in an inaccessible place. The average fireman or operating engineer takes pride in the mechanical equipment in his charge, and ordinarily

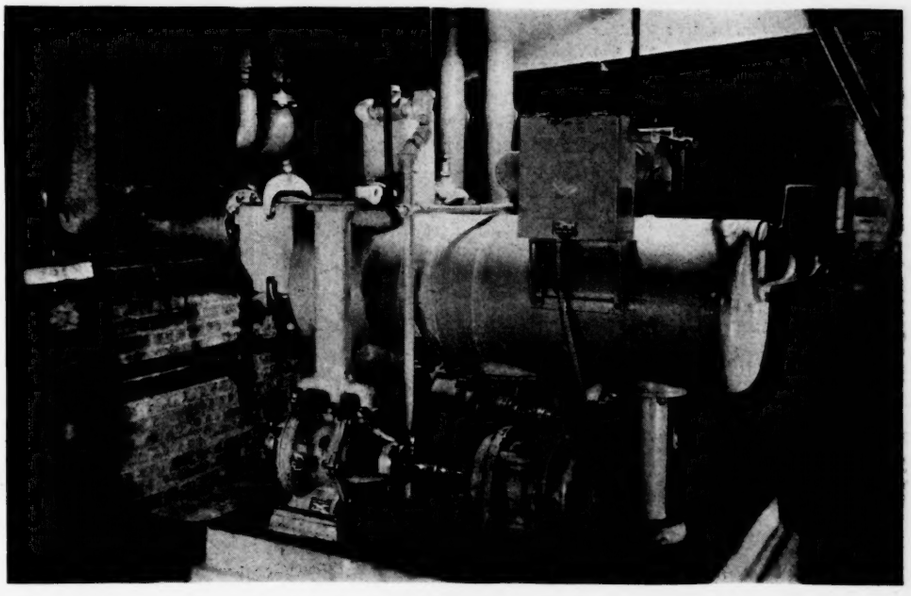


Fig. 3. A satisfactory installation except that individual gate valves in the return lines were omitted. Such valves would have aided in locating leaky traps

will keep it clean and in good condition. This means longer service from the equipment, and is worth taking into account when considering the location of a pump.

Vacuum heating pumps should be placed far enough from the boiler so as to be free from ash dust, yet one often sees pumps right next to ash pits with the motor windings packed with dust. This condition often results in the overheating and burning out of motors. Practically all manufacturers employ a protective thermal relay with their electrical equipment, and often the heat from the boiler will cause the motor to stop or cut out only to start again after the boiler room temperature has fallen. Many service calls have been made from this one complaint.

Some architects have a 4-in. brick wall built to provide a space which can be converted into a pump room and which can be kept clean and free from dust.

Preliminary planning in the proper placing of a pump and its piping may be the means of effecting a saving in both material and labor. Some heating contractors regard preliminary planning, followed up with properly prepared plans, as unproductive and unnecessary. They accept plans or layouts without considering the feasibility of re-arranging details. Often the heating contract is the last one to be let, long after the building is started, and it is found that the whole heating system must be installed under difficulties. Preliminary planning will do much to prevent such conditions.

A vacuum pump should be placed, and the piping so arranged, that condensate will flow into the receiver tank suction connection by gravity. This is far better than resorting to lifts. While lift connections may be necessary at times, a pump operating on a suction lift cannot be expected to pump effectively with condensate at 180° to 200°. Such a temperature in the return lines may be due to leaky traps, and vaporization under vacuum will cause air and vapor binding in the piping.

When one or more pumps are to be installed it is best to allow sufficient space for a man to get around them with ease. See Fig. 5.

In laying out a pump or boiler room with the view of determining the best possible location for each piece of apparatus, one method is to make a plan of that portion of the building on a large scale and to use scaled templates made of heavy drawing paper. These templates can include the space necessary for removing tie bolts or boiler flues. The templates may be shifted from place to place on the plan until the best location for each piece of apparatus is determined.

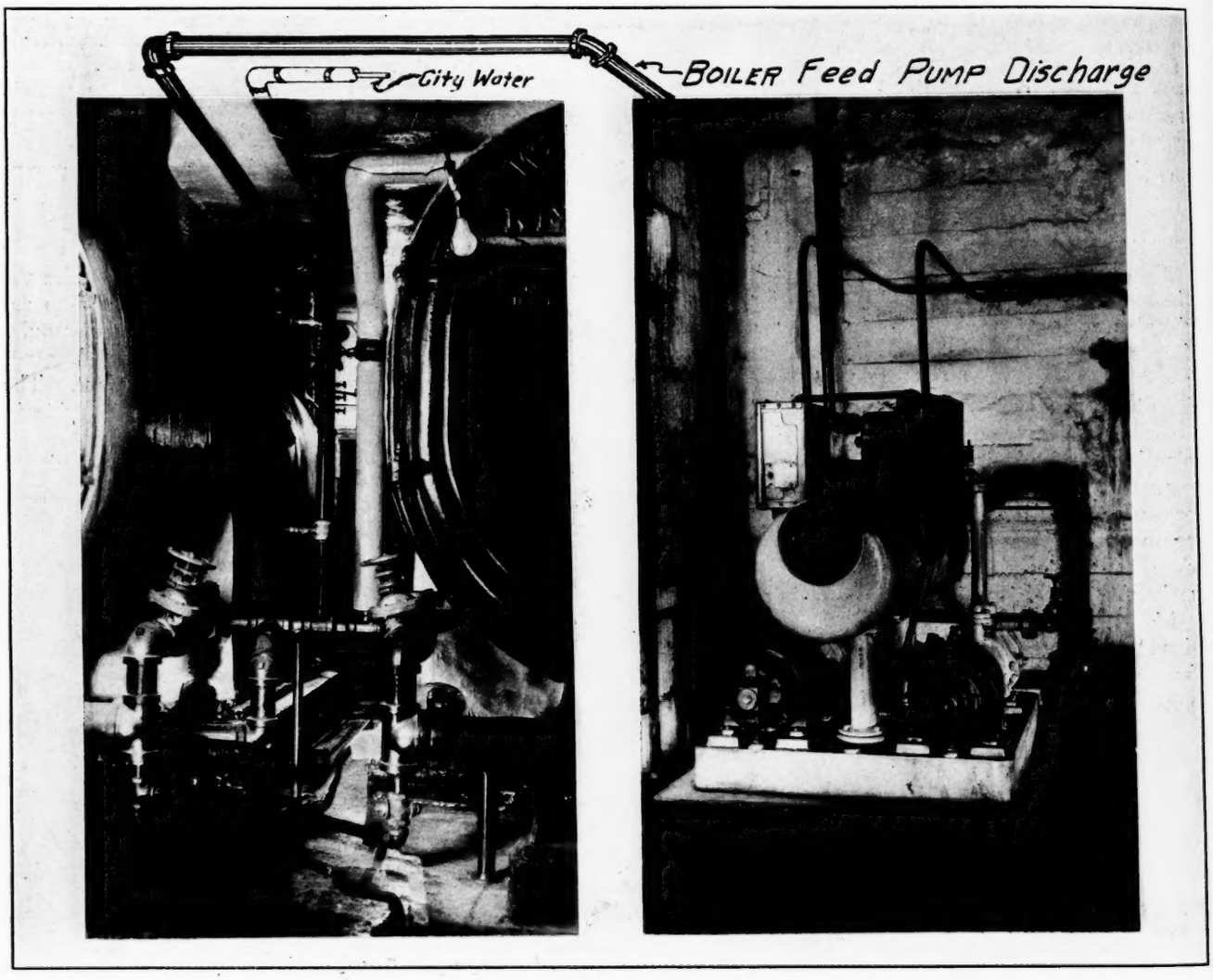


Fig. 4. A duplex vacuum heating pump. The condensate is lifted at the lift connection at the wall

Drains to the sewer should be conveniently located, for there will be some water leakage around a pump from time to time. Convenient drains also facilitate draining the pump receivers and the cleaning of dirt strainers. Often makeshift drain lines are resorted to, with piping run across the floor to the nearest drain. This detracts from the appearance of installation and creates a safety hazard. A permanent electric light over the pump is to be preferred to an extension cord hung by a rope.

Pump Foundation

Bed plates holding the vacuum pump in place should be level and erected on a suitable concrete foundation from 3 in. to 2 ft. in height, depending on the piping details to be accommodated and the possibility of water floods rising to the level of the motor. If the return piping and discharge lines are overhead, an extra height to the foundation will make the pump more accessible for the operating engineer. After the pump location has been decided on a foundation plan should be prepared. See Fig. 2.

It is far better to have the bolts set in firmly when the foundation is poured than to have the holes drilled later. Anchor bolts should be placed in a casing consisting of a 1½ in. or 2 in. pipe nipple. The casing will permit lateral movement in case of inaccuracies in locating the bolts. The length of the bolt should be such so as to allow 1 in. between the top of the foundation and the bottom of the pump bed plate. Wedges are employed to level the pump and a space is left to permit grouting. Bolt casings do not have to extend through the grout. The casing should not be filled with cement, but with sand, rags or paper.

The foundation should have a final coat of cement rubbed in with a trowel before it is dry. The upper part of the foundation should be painted to protect it from deteriorating and being affected by oil. See Figs. 1 and 5.

Piping at the Pump

When duplicate pump installations are made, each pump should be so connected as to operate independently of the other. If the layout of the plant permits by-passing of the return piping direct with the boiler, this is always recommended.

Thought should be given to safety. Piping and valve stems are often so assembled that it is possible for a person to be hurt or to tear his clothing as he passes by.

A boiler compound reservoir should be made up and installed in the discharge boiler feed line.

When a system has more than one return main leading to a header, a gate valve should be installed in each line before it is connected to the suction header. This is often omitted on new work, with the result that it is impossible to determine what part of a system may be leaking steam or air. Trouble-finding is greatly facilitated by shutting off each valve and then watching the performance of each return on the vacuum gauge at the pump and the thermometer that should be installed in every vacuum return system.

An equalizer pipe line should be connected between the boiler steam header and the end of the returns of the vacuum heating system. This is necessary because of the natural vacuum which forms during night operations or any time when the fire in the boiler is low. The equalizer pipe will prevent the condensation and air from being held back in the system.

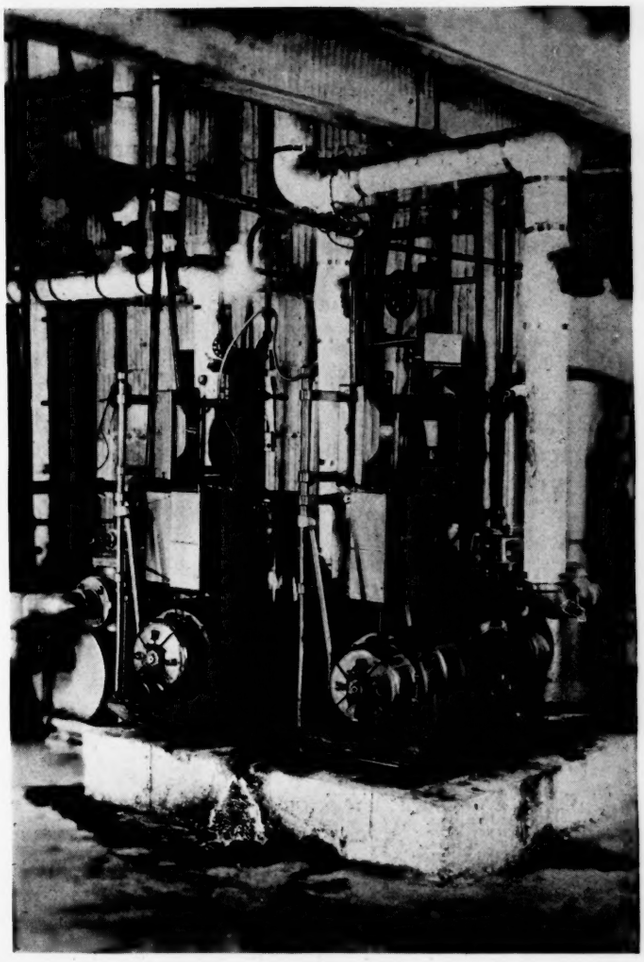


Fig. 5. An extremely crowded installation. The parts are inaccessible in case repairs are to be made

The equalizing line may be made up of ½ in. or ¾ in. pipe, the equalizing feature being accomplished with a swing check valve in a 4 in. or 6 in. water seal. Instead, a thermostatic return trap can be used, which equalizes any difference in pressure between the supply when there is no steam pressure.

When a water pocket is unavoidable, as for example when a suction lift is located in the return near the pump, the vacuum regulator should be connected with the return piping. This is accomplished by connecting the vacuum regulator to the top of the return header, about 15 ft. away from the pump. Such a connection can be used to prevent continual starting and stopping of the pump, which may occur when slugs of water are lifted.

Tightness of a vacuum heating system can be tested by setting up the vacuum regulator at the pump so it will create 10 in. or 15 in. of vacuum and maintain it for an hour or so. A new system is not apt to hold as high a vacuum as it will after it is in use the second season, due to the sealing of the pipe joints by corrosion.

Operation of a New System

In starting a new vacuum heating system it is important that the system be operated without the thermostatic elements in the traps, and that temporary covers be used instead. Condensate should be drained into the sewer direct for some time—perhaps for a month. A high steam pressure at the boiler is desirable to force from the system all the pipe thread shavings, sand, and oil.

While this plan will remove most of the dirt, it is not to be expected that all the foreign matter will be held back by the strainer at the pump suction. During the first year or two there is likely to be a continual flow of particles, and special provision for draining this dirt is desirable. Temporary extra-long dirt pockets with a large capacity, and with a valve at the end, are useful. After the first year these can be removed and permanent dirt pockets installed.

Where this has not been done, small particles of abrasive material are known to have passed the strainer, and to have worn away the impellers and cut the shaft. The greatest wear on a vacuum pump usually occurs the first year or two.

Instruments in the Return Line

To have a constant check on the performance of the vacuum return, it is necessary that suitable gauges and thermometers be installed. When gauges or thermometers are not specified, it is a good plan for the contractor to install fittings with plugged outlets for their future installation. He himself may find them useful to locate trouble before he leaves the job.

While a condensation meter is not essential for the successful operation of a vacuum heating system, its installation is insisted on by some engineers. It is connected in the discharge line of the pump, by-passed, and the supply water is so connected as to permit it to pass through the meter.

On most vacuum heating systems the only vacuum gauge will be the one on the pump. It usually remains there until it is completely out of order. It is advisable to have another gauge near the pump connected at the return suction header in order that full dependence need not be placed on one gauge. A compound gauge, instead of a vacuum gauge, is desirable if there is a possibility of steam pressure getting into the return.

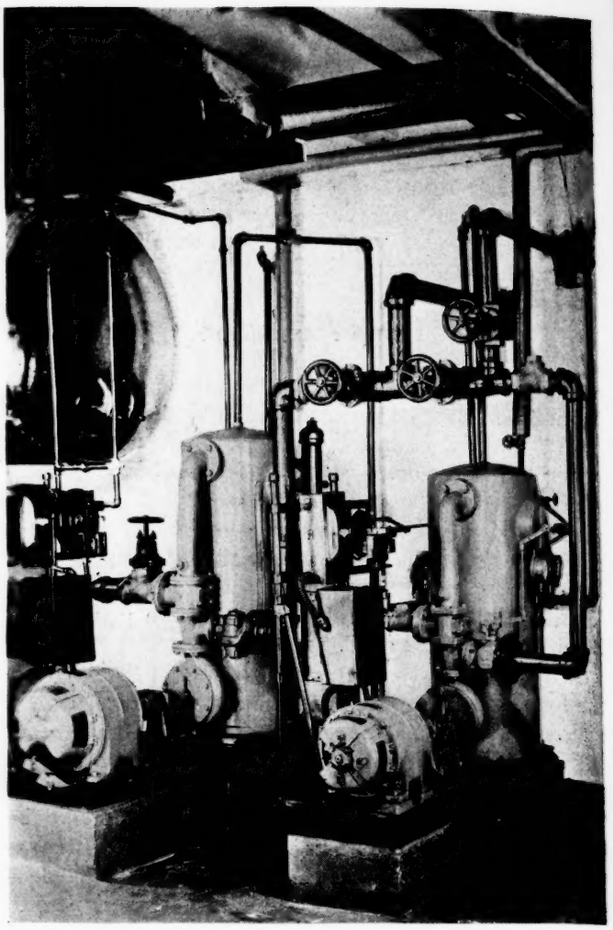


Fig. 6. An exceptionally good installation. The piping is neat, there is sufficient space around the pumps, the foundations are sufficiently high, and a drain has been provided between the pumps

A good grade thermometer should be installed in the return header. It is essential that the bulb and stem of the thermometer be submerged in the returning condensate. An industrial type thermometer should be used instead of relying on a cheap instrument. Use of a thermometer helps to detect leaky traps. A sudden rise of condensate temperature may be due to a piece of scale lodged under the seat of one of the traps. Prompt location and correction of one such case will more than pay for such an instrument in labor and fuel saved.

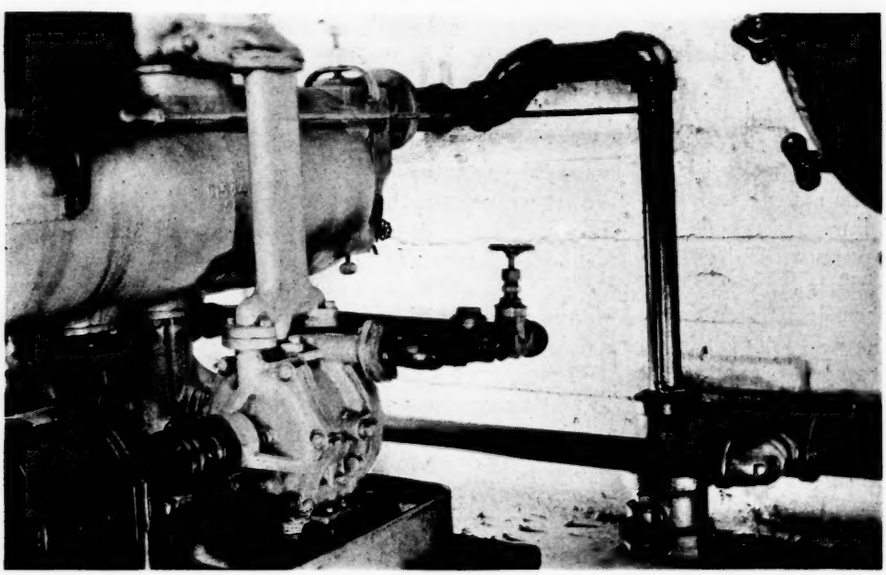


Fig. 7. A typical suction lift connection. Note the small pipe connected at the top of the return which is joined to the vacuum regulator of the pump

Alignment Charts for Solving Kent's Chimney Equation

By J. N. ARNOLD†

THE proportions of a chimney may be rapidly determined with an alignment chart solution of Kent's chimney equation. Two of the three quantities, number of square feet of radiation (or boiler horsepower), height of chimney and area of chimney must be known, and the other quantity is determined from the chart. A scale is also included for converting diameter to area. Two charts are shown, with overlapping ranges—one for the smaller chimneys with capacities in square feet of radiation, and one for large chimneys, in boiler horsepower. The symbols are as follows:

S = rated capacity of radiation (in square feet) of boilers served by the chimney.

H = height of chimney, feet.

A = inside area of chimney, square feet.

D = inside diameter of chimney, inches.

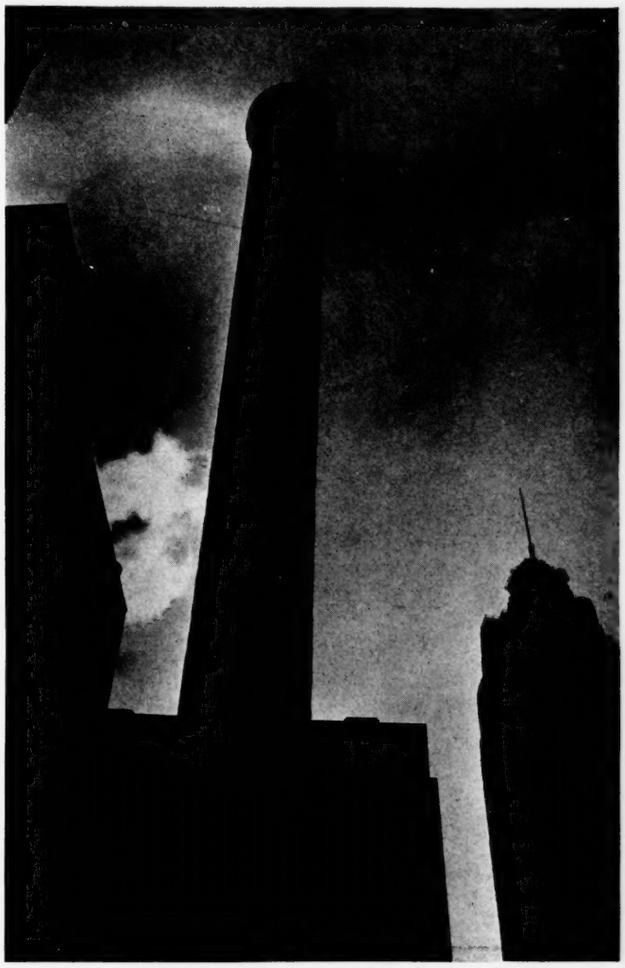
To demonstrate the manner of using Fig. 1 an example will be solved.

Example: What diameter chimney 150 ft. high would be required for a heating plant with a boiler having a rated capacity of 55,000 sq. ft. of radiation?

Solution: Through 55,000 on the S scale and 150 on the H scale draw a straight line to the D scale. On the D scale read the diameter of the chimney as 46.4 in., and on the A scale the area 11.75 sq. ft.

In a similar manner, and even more rapidly, the height of chimney can be determined when the diameter of the chimney and the boiler horsepower are given.

To determine the most economical proportions for a chimney to serve a given boiler plant several different values of the diameter, D, may be assumed and the corresponding height determined by means of the chart. From estimates of the cost of each of the several trial solutions the most economical can be selected.



To illustrate the use of Fig. 2, assume a chimney 71 in. diameter and 150 ft. high. What boiler horsepower can be accommodated?

Locate 71 on Scale D and on the other side of the line read the area as 27.5 sq. ft. Locate 27.5 on the A scale to the right of the chart and connect the same points (27.5) on the two A scales. Through the intersection of the line with the Q-axis, draw a line through the point 125 on the H scale. On the H.P. Scale read 1000 hp.

Similarly, if the horsepower and height are known, draw a line through these points on the H and H.P. scales till it intersects with the Q-axis. Adjust the straight-edge around this point on the Q-axis until identical readings can be found on the two A scales.

† Instructor, Engineering Drawing, Purdue University.
Photo by Ewing Galloway

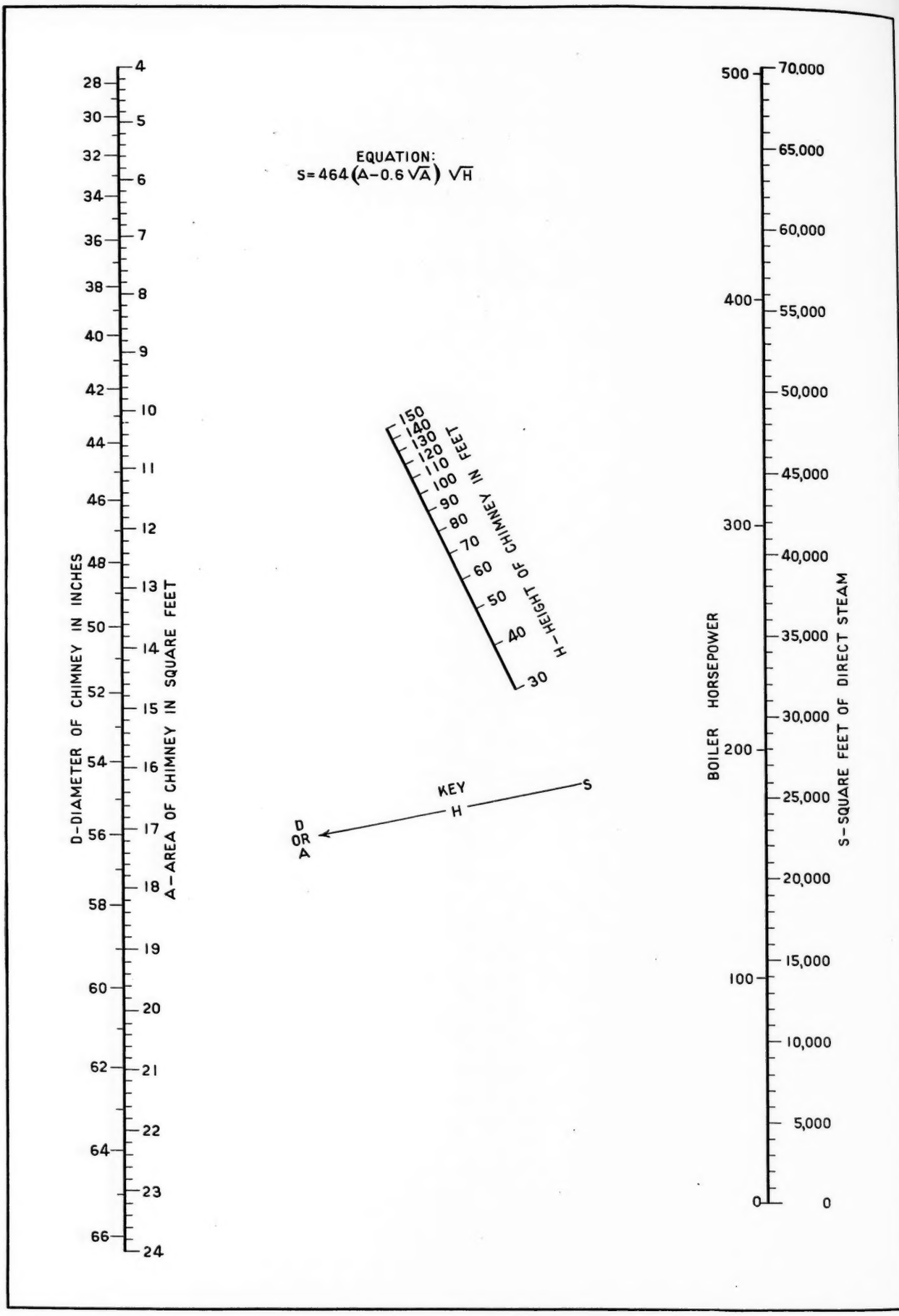


Fig. 1. Nomographic chart for determining chimney sizes up to 150 ft.

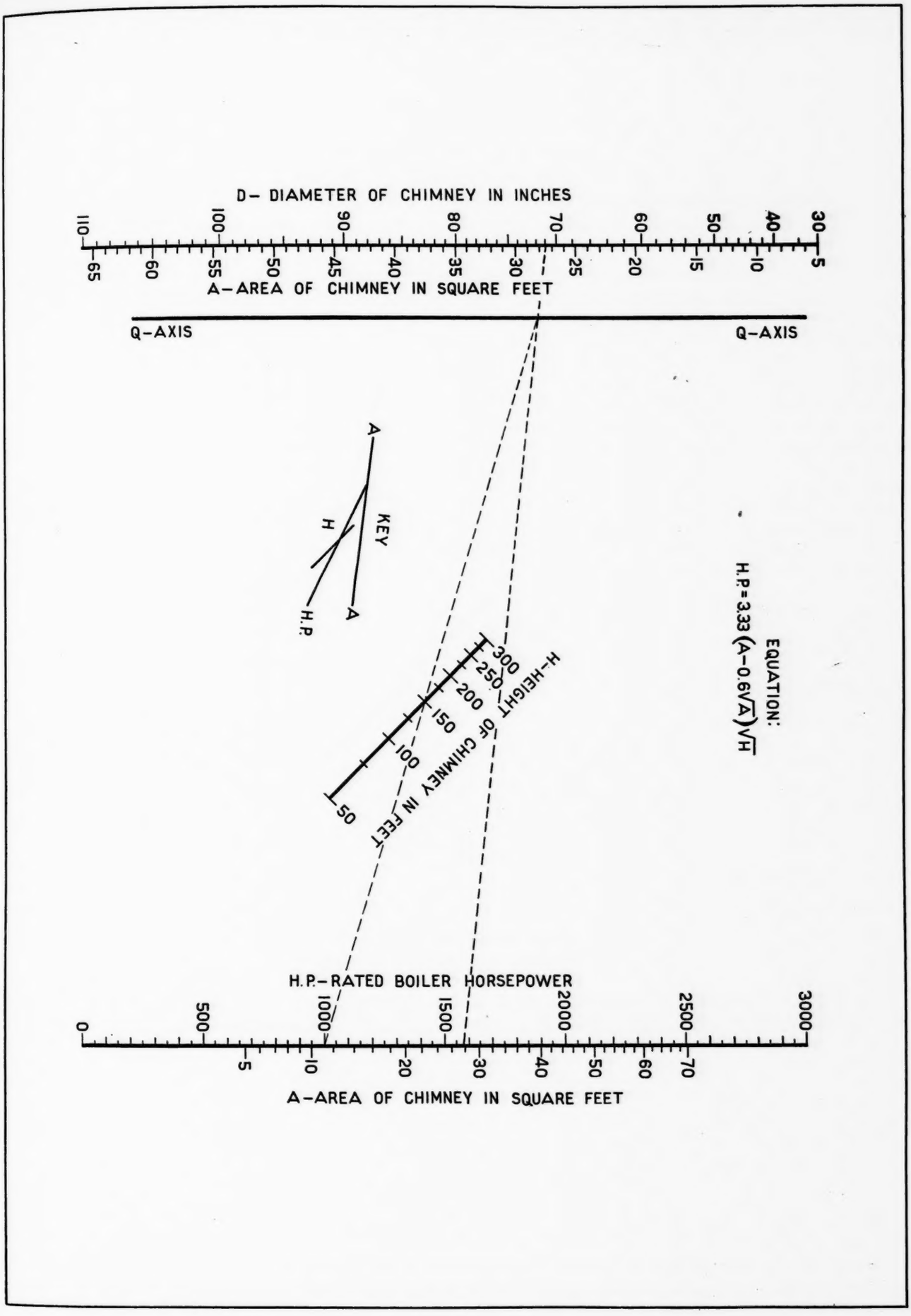


Fig. 2. Chart for determining chimney sizes for large heating plants and small power plants.

The Labus Psychrometric Chart

By C. O. MACKEY†

IN previous issues of this publication,[1] the principles of the construction and use of several psychrometric charts in common use were reviewed. A psychrometric chart, different in construction, but having much the same field of application as the Bulkeley, Carrier, and Hill charts, has since come to the attention of the author. This chart is copyrighted by O. A. Labus.

In fundamental principle, the Labus chart (Fig. 2) is a network chart with a primary coordinate system of relative humidity plotted against wet bulb temperature, both on uniform scales. On this primary coordinate system, lines of constant absolute humidity (in grains per pound of dry air) and lines of constant dry bulb temperature are located, in accordance with accepted psychrometric principles, for the single barometric pressure of 29.92 in. of mercury. The lines of constant absolute humidity are also lines of constant dew point, and may be labeled with values of absolute humidity and dew point, as in the skeleton chart of Fig. 2, although, in the original chart, an auxiliary scale is carried along the abscissa to give the absolute humidity. A scale for the secondary variable, "total heat," is placed adjacent to the scale of wet bulb temperature, upon which it depends. In form, then, the Labus chart is quite similar to the Hill chart, for on the latter, dew point temperature, with dependent secondary variables, is plotted against dry bulb temperature, and lines of constant wet bulb temperature and constant relative humidity are drawn on this coordinate system.

A numerical example, illustrating the use of the chart, is now solved, and the various state points entering this solution are shown on the skeleton Labus chart of Fig. 1.

Assume the state of the outside air to be shown by 1 on the diagram, i.e., with a dry bulb temperature of $t_1 = 38°$ and a wet bulb temperature of $t^1_1 = 36°$. The total heat of this air may be read directly on the auxiliary scale as $th_1 = 13.4$ B.t.u. per lb. of dry air, since $t^1_1 = 36$. The absolute humidity may be read as $A\,H_1 = 27.6$ grains per lb. of dry air.

Assume that the air is to be discharged from the re-

† Assistant Professor of Heat Power Engineering, Cornell University.

1 "A Review of Psychrometric Charts," Part I, June, 1931, page 50; Part II, July, 1931, page 64; Part III, August, 1931, page 59.

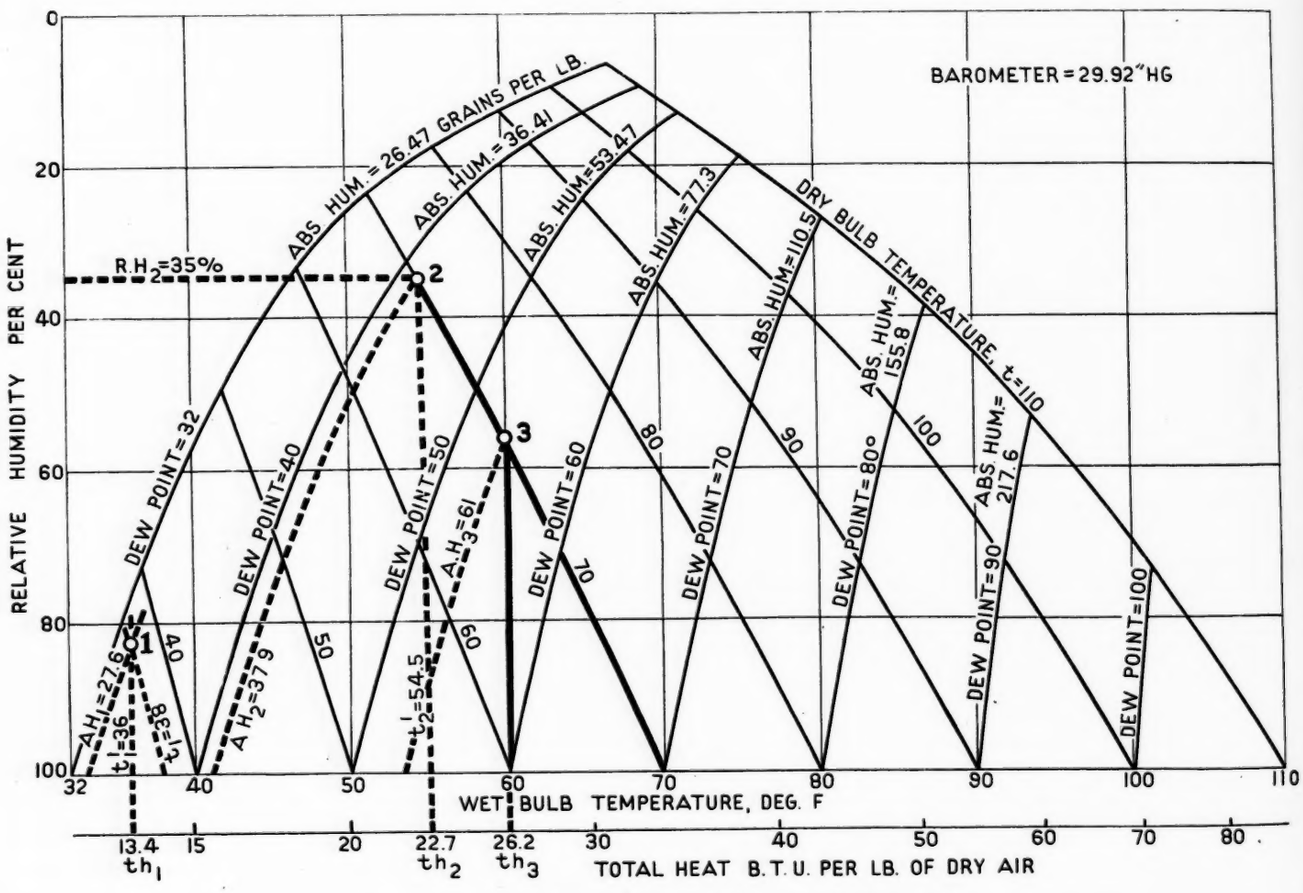
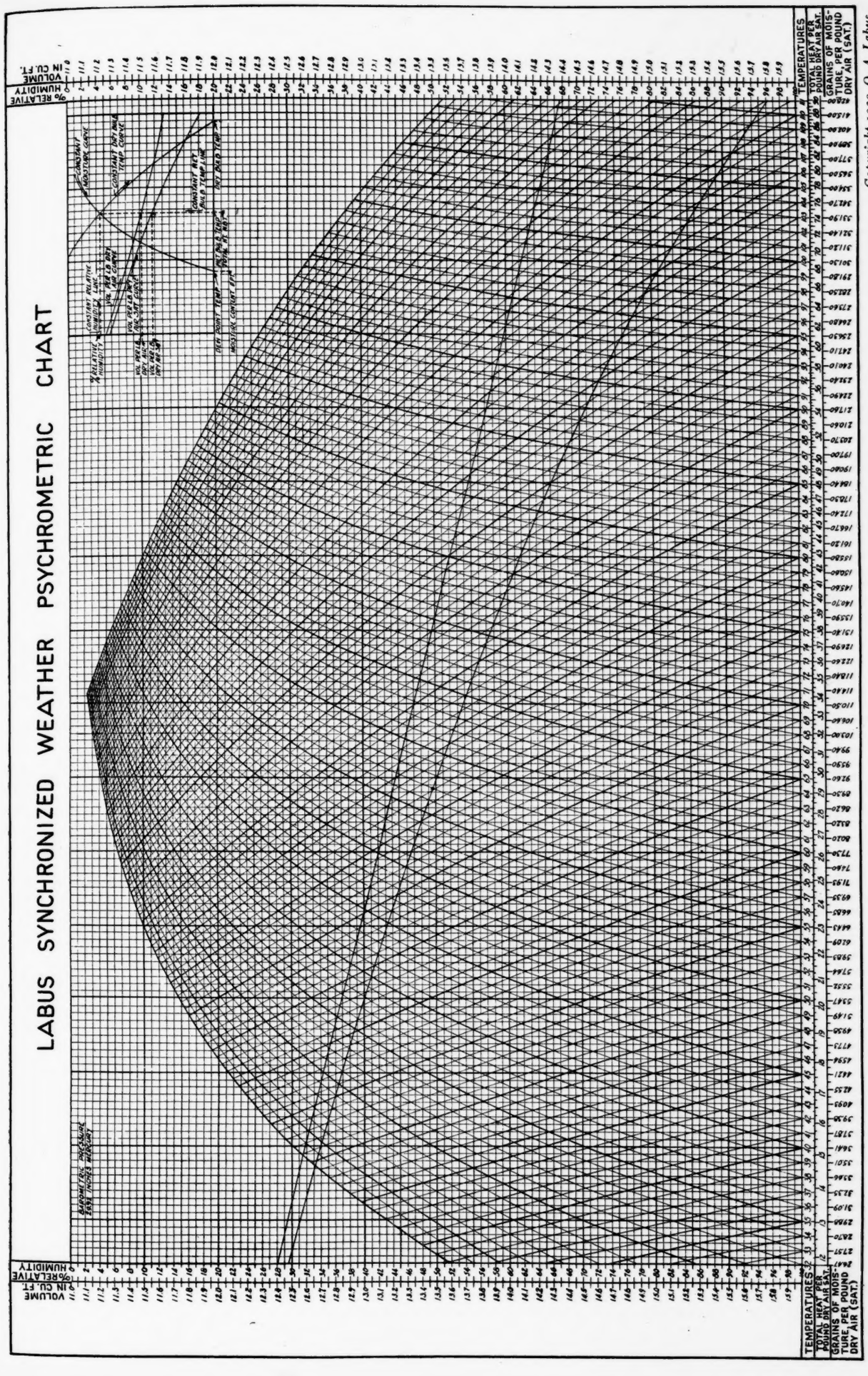


Fig. 1. Solution of numerical example on the Labus chart


LABUS SYNCHRONIZED WEATHER PSYCHROMETRIC CHART
% RELATIVE HUMIDITY
VOLUME IN CU. FT.
TEMPERATURES
TOTAL HEAT PER POUND DRY AIR SAT.
GRAINS OF MOISTURE PER POUND DRY AIR (SAT.)
CONSTANT MOISTURE CURVE
CONSTANT DRY BULB TEMP. CURVE
CONSTANT WET BULB TEMP. LINE
DRY BULB TEMP.
MOISTURE CONTENT LINE
BAROMETRIC PRESSURE 29.92 INCHES MERCURY

Copyright 1930, O. A. Labus

Fig. 2. The Labus chart, designed for use in solving air conditioning problems

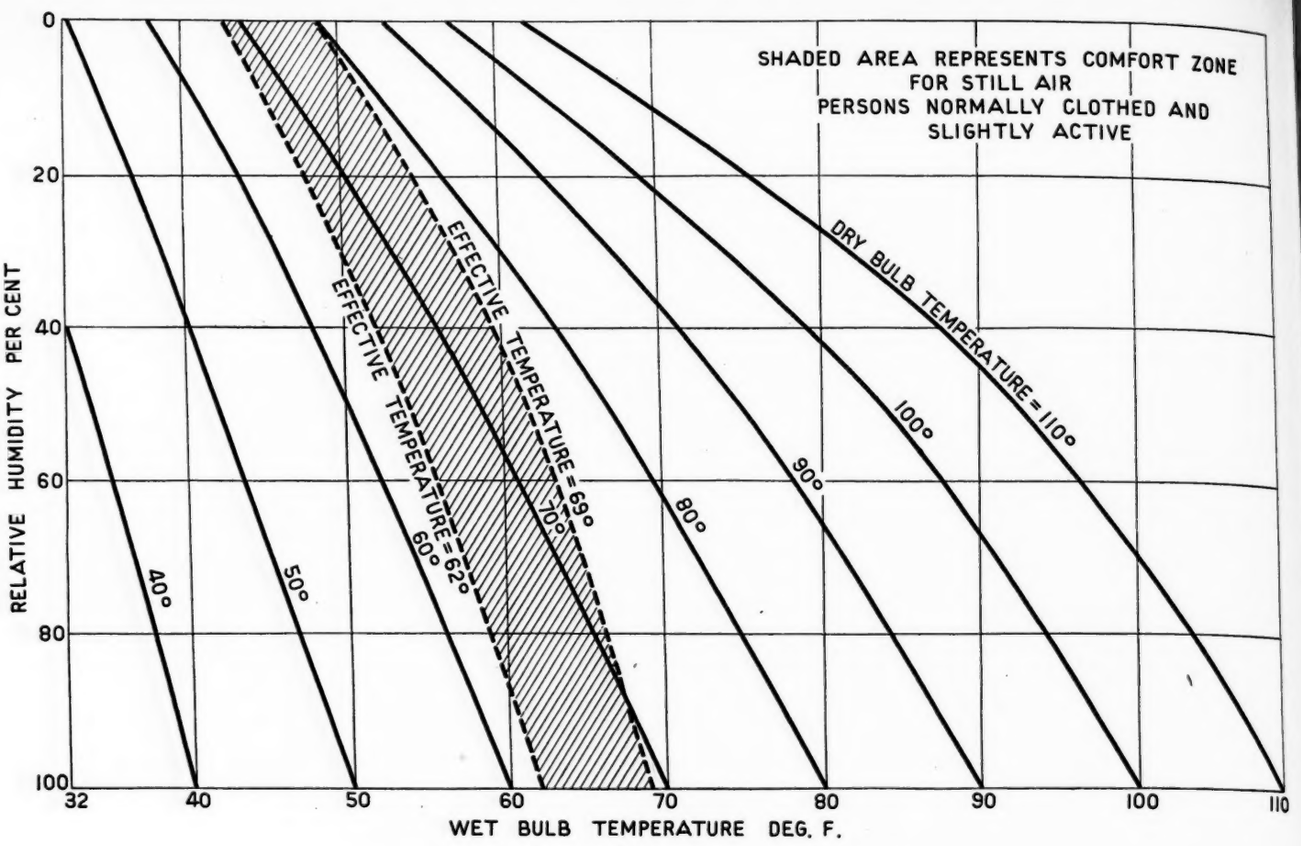


Fig. 3. Comfort zone boundaries on Labus chart

heater of the air conditioning system in state 2 with a dry bulb temperature of $t_2 = 70$ and a relative humidity of R. $H_2 = 35\%$. Then the state point, 2, is located as shown. At this state point, the wet bulb temperature is $t^1_2 = 54.5°$, the total heat, $th_2 = 22.7$ B.t.u. per lb. of dry air, the dew point is 41° and the absolute humidity is A $H_2 = 37.9$ grains per lb. of dry air.

If the recirculated air be in state 3, with a dry bulb temperature, $t_3 = 70$, and a wet bulb, $t^1_3 = 60°$, the state 3 is located as shown. In this state, $th_3 = 26.2$ B.t.u. per lb. of dry air, and A $H_3 = 61.0$ grains per lb.

If the reheater is discharging 1000 lb. of dry air per min., and if this is made up of 70% outside air, then the preheater and reheater must add to the air a total quantity of heat equal, approximately, to

$$Q = (1000 \times 22.7) - (700 \times 13.4) - (300 \times 26.2) = 5{,}460 \text{ B.t.u. per min.}$$

Also, the humidifying sprays must supply moisture at the rate of

$$W = (1000 \times 37.9) - (700 \times 27.6) - (300 \times 61) = 280 \text{ lb. per min.}$$

In addition to, but distinct from, the main coordinate system of the chart, the specific volume of bone dry air and the specific volume of air saturated with vapor, in cubic feet per lb. of dry air, are plotted against dry bulb temperature. On the basis of relative humidity, it is possible to interpolate between these lines and secure the humid volume.

In Fig. 3, the two lines of constant effective temperature bounding the comfort zone, for still air and for persons slightly active and normally clothed, are shown in their relation to the lines of constant dry and wet bulb temperature on the Labus chart.

Tuberculin Manufacture Requires Air Conditioning

CAREFULLY controlled atmospheric conditions are necessary in the laboratories of the United States Department of Agriculture where ophthalmic discs used in tuberculin production for research in the combating of tuberculosis in cattle are made.

Liquid tuberculin has the property of absorbing moisture from the air under ordinary room conditions, making the liquid worthless. Government engineers finally developed a system consisting of glass cabinets connected to a refrigerating machine so that the moisture is deposited as ice on the metal coils. The glass cases contain the dried air, and it is in these cases that the ophthalmic discs are mixed and formed. The air after being cooled to remove the moisture, is heated electrically to make the conditions comfortable for the scientists who stand on the outside of cabinets with their hands inserted through carefully designed openings.

Prior to the development of the dehumidifying system it was possible to manufacture the discs only four weeks out of the year, and since the annual requirements are in the neighborhood of three million discs, it was physically impossible to meet the demand.

Use of the Labus Chart for Air Conditioning Operation

By K. P. BRACE†

THE success of any air conditioning job usually depends not only upon correct design and installation, but upon intelligent operation as well. In most cases, however, the man who is called upon to operate such a system is a mechanic or licensed stationary engineer, who, although well versed in his particular line, is not familiar with the theories or principles of air conditioning. Since the conditioning of air is a rather intangible process, difficult to visualize without a great deal of thought and study, the average mechanical man who is picked to operate an air conditioning system usually finds it exceedingly difficult to comprehend in its entirety the system he is operating, even though he is thoroughly familiar with the intricacies and idiosyncrasies of component parts such as fans, pumps, motors, and refrigerating machines.

In many cases, the only source of information available to the engineer is the operating instructions issued by the manufacturer of the air conditioning equipment, which are usually incomplete and difficult to understand.

The chart shown in Fig. 1[1] was designed to enable the operating engineer of a system laid out as shown in Fig. 2 to tell at a glance just what parts of the system should be run for the most economical results with varying outside conditions.

Fig. 1 is the Labus psychrometric chart[2] divided into ten numbered zones containing all weather conditions (except extremely cold conditions) that are likely to be encountered in air conditioning work.

Table 1 shows just what should be done in each zone.

Naturally, the zones and the instructions in the table will vary with location, design of the system, ordinances or requirements as to outside air and inside air requirements, but the idea is very clear and easy to understand; hence, it should be helpful to any engineer operating an air conditioning system.

† Air Conditioning Engineer, National Carbonic Machinery Co., Chicago.

1 The chart used was a full-sized chart similar to that shown on page 41. Fig. 1 is a skeleton only, showing the zone boundaries.

2 See preceding article, The Labus Psychrometric Chart.

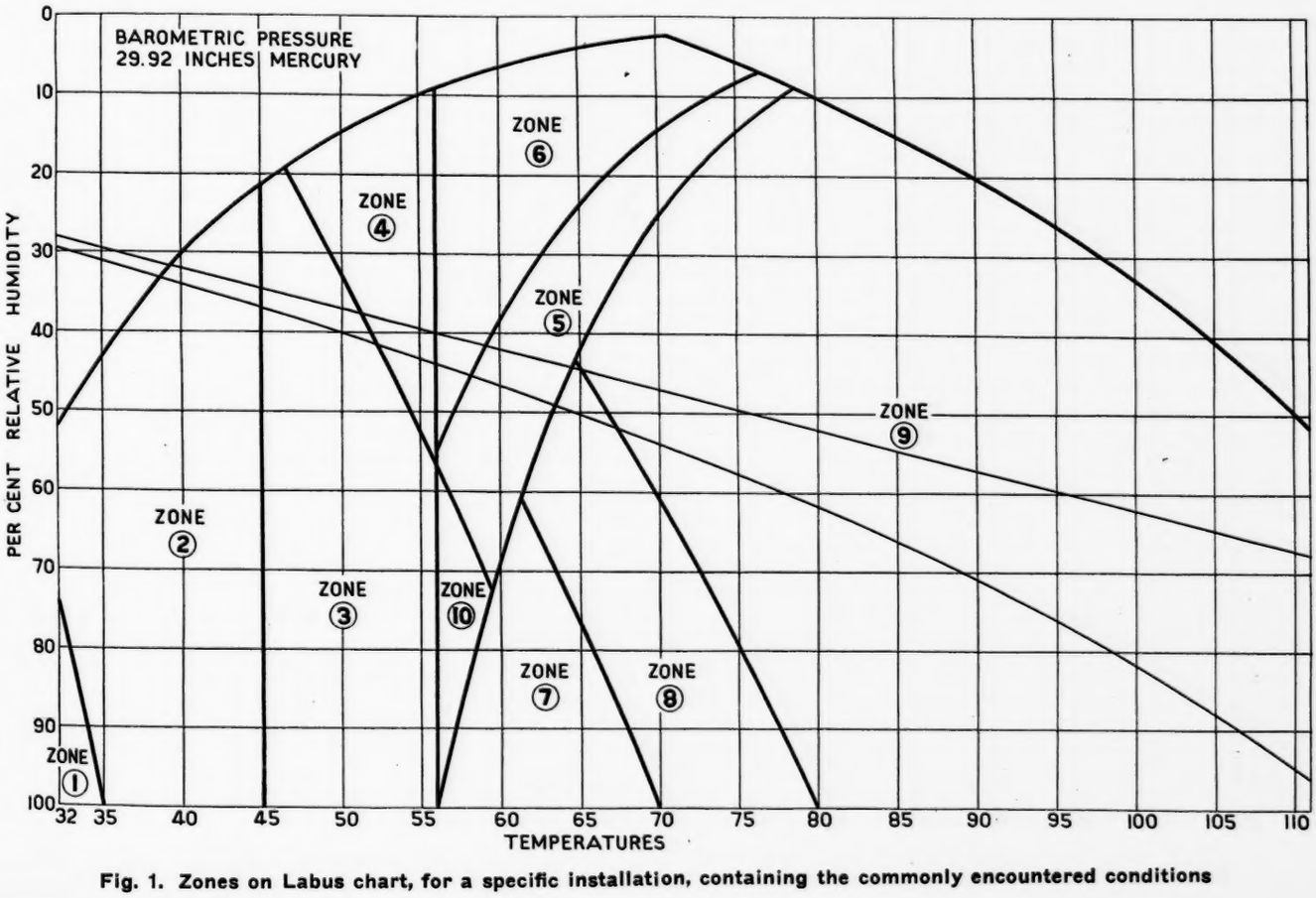


Fig. 1. Zones on Labus chart, for a specific installation, containing the commonly encountered conditions

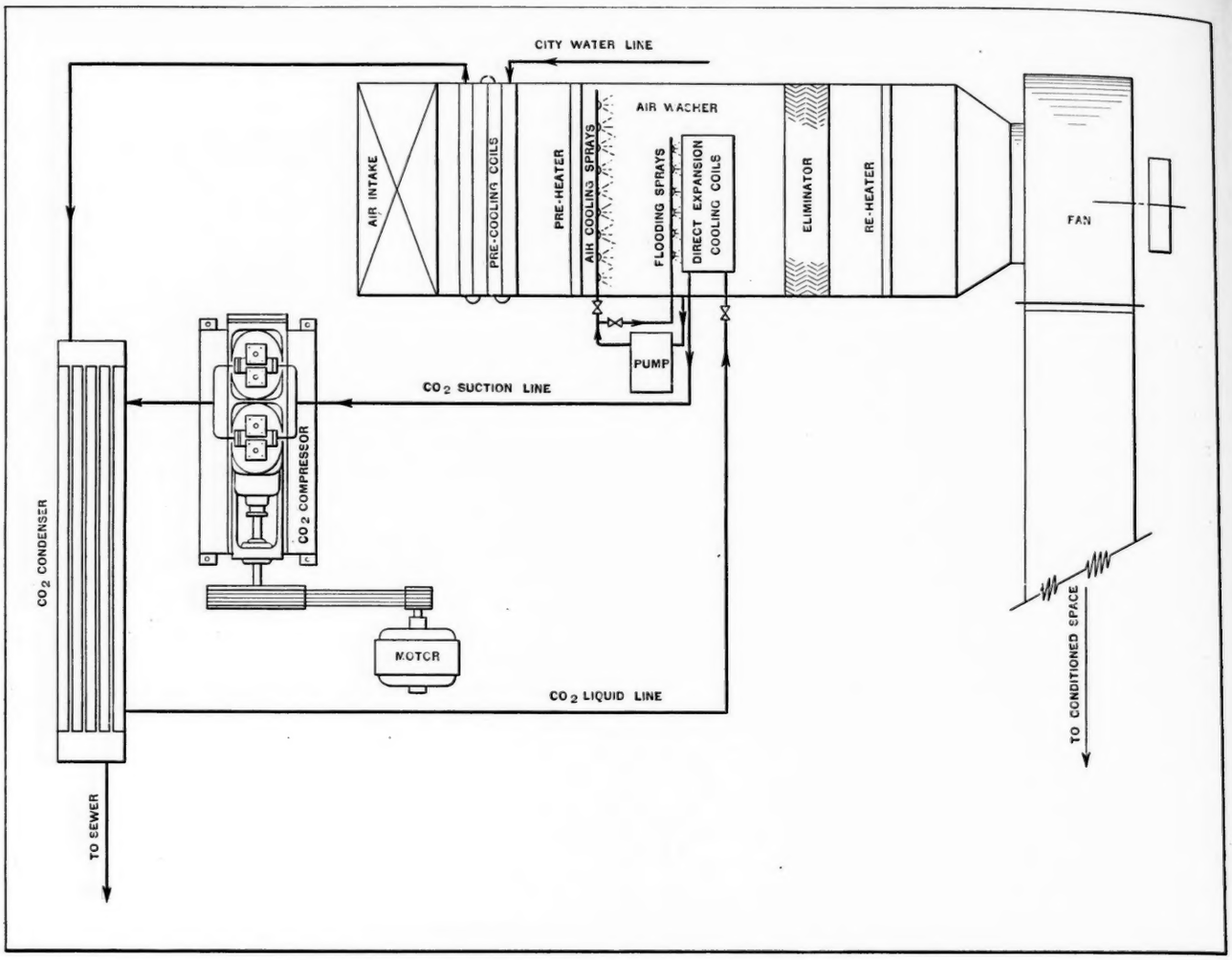


Fig. 2. System for which chart, Fig. 1, was designed

The system (Fig. 2) for which this particular chart was designed consisted of the following parts:

1. Air intake in which the proportion of outside to return air can be varied between the limits of 25% outside air to 100% outside air.
2. Eight rows of extended surface precooling coils. These coils are supplied with 58° water from the city mains and are capable of cooling the air which passes through them from 95° to 65° when the entering dew point was 65° or below. The city water which passes through the precooling coils is used in a shell and tube condenser for condensing the CO_2 refrigerant.
3. One row of extended surface preheating coils.
4. One bank of mist type sprays for cooling and saturating the air.
5. One bank of flooding type sprays for sluicing the direct expansion coils.
6. Extended surface CO_2 direct expansion coils for cooling the water in the dehumidifiers. These direct expansion coils do considerable cooling and dehumidifying when not flooded with water.
7. One row of extended surface reheaters.
8. Fan.

This chart is based on an inside condition of 70° and 35% relative humidity during the winter and a maximum inside summer condition of 80° and 48% relative humidity, allowing for a 2° rise in the dew

TABLE 1
OPERATING INFORMATION

Zone	Precoolers	Preheaters	Sprays	Flooders	Expansion Coils	Reheaters	Refrigeration Machine	Per Cent Outside Air
1	Off	On	On	Off	Off	On	Off	50 to 25
2	Off	On	On	Off	Off	On	Off	50 to 100
3	Off	Off	On	Off	Off	On	Off	100
4	Off	Off	On	Off	Off	On	Off	100
5	On	Off	Off	Off	Off	Off	Off	100
6	On	Off	On	Off	Off	Off	Off	100
7	On	Off	Off	Off	On	Off	On	50
8	On	Off	Off	On	On	Off	On	50
9	On	Off	On	On	On	Off	On	50 to 25
10	Off	Off	Off	Off	Off	On	Off	100

point, which means that the dew point temperature of the air leaving the dehumidifier should be 56°.

Zone 1 is a winter condition requiring preheaters for humidity control, sprays for saturation, and reheaters for temperature control. In Zone 1, it is economical to cut the outside air to a minimum.

Zone 2 also requires full heating, but in this zone, temperatures are milder and more outside air can be used.

Zone 3 requires reheating, but no preheating, because in this zone the outside wet bulb temperature is between 45° and 56°. This being the case, the air will be saturated somewhere between 45° and 56°, which will result in relative humidity between 40% and 60% when an inside temperature of 70° is maintained.

In Zone 4, dry bulb temperatures are above 65° and wet bulb temperatures are below 56°. In this zone, when temperatures are 70° or over, the necessary cooling can be accomplished by running the air washer only. When temperatures are below 70°, the reheaters may be required, depending upon the amount of heat generated within the building. All outside air should be used in Zone 4.

In Zone 5, dew point temperatures are below 56° and dry bulb temperatures are anything over 65°. In this zone, the most economical practice is to run the precoolers only. If the air washer is turned on here, humidity will be too high and dehumidification will be required.

In Zone 6, the dew point temperatures are below 49.5°, which means that if the entering air is cooled by means of precoolers to 65°, the resulting wet bulb will be below 56°; hence, the air washers may be run in connection with the precoolers without causing the humidity to be too high.

Zone 7, with dew points over 56° and dry bulb temperatures below 70°, requires dehumidification, but little or no cooling. This can best be accomplished by running the refrigerating machine and the direct expansion coils with no sprays or flooders on. In a system containing a by-pass, the by-pass damper should be wide open in this zone.

Zone 8 requires considerable dehumidification and a moderate amount of cooling. Hence, the most economical practice is to run the refrigerating machine, direct expansion coils, and flooders, which, producing a coarse spray, do not completely saturate the air. In this zone, if a by-pass were used, the damper should be approximately half open, depending, of course, upon how high the entering dew point is above 56°.

Zone 9 is the maximum cooling zone, in which the entire system should be run at full capacity.

Zone 10 applies only to a few rare days on which nothing is required except a small amount of reheating.

Form for After-Hours Heat in Office Building

USE of a standard form filled in by tenants who desire heat in their quarters after regular hours has resulted in savings of 50% in steam consumption during off-hours in office buildings, according to Alfred Pomn, of Detroit, writing in a recent issue of *Skyscraper Management*.

The management supplies the tenants with the special heat request forms, which must be filled out in triplicate and left with the desk clerk previous to the time that the extra heat service is required. Copies are filed with the engineer and with the office records.

The forms provide space for filling in the tenant's name, the floor space occupied, name of the individual who is requesting the heat, date, and time of day when the extra heat is desired. A floor plan of the building is included, and the tenant checks the room to be occupied. There is also space to be filled out by the management, including the name of the recipient, the name of the engineer on duty, the date and time the request was received, and verification of the date and time for which the heat was requested.

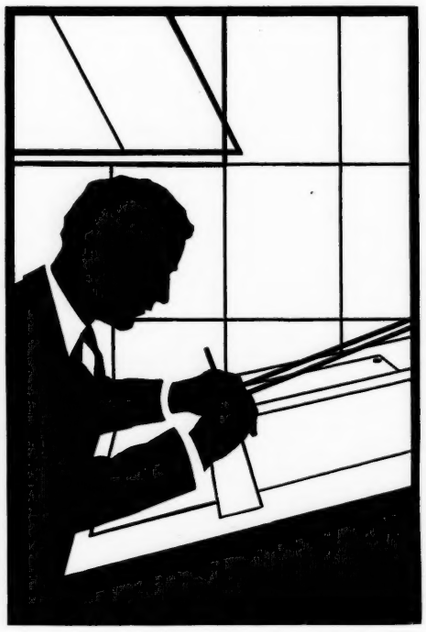
The building in which this system has been put in practice is equipped with pneumatic control steam valves, so that the risers supplying certain portions of the building may be turned on only when steam is needed in these zones. The valves are operated from a central control panel in the main engine room, and the operator in charge can furnish heat to any particular section of the building by turning on the valve controlling that zone, making it unnecessary for him to leave the engine room.

Where such a system is not used, extra heat is provided on verbal request, and in the majority of cases an appreciable amount of steam is wasted due to tne heating of unoccupied space, in addition to the rooms being used. If the time is not stipulated, waste also results from heating the office when no one is using it. With the time of occupancy definitely designated, the heat is turned off promptly when the tenant leaves.

The management of the building following this policy has noted that a number of the tenants requesting extra heat, withdraw the request when they find that filling out the form is necessary, saying that they only intend to work a short time, and will not bother. It is estimated that the system has brought about a reduction of heating costs for evenings, Sundays, and holidays, of approximately 50%, a saving of $300 per month.

Some Suggestions for Specification Writers

By CHARLES A. CARPENTER



THE main hope for the speedy return of good times depends upon the building trades. Although authorities differ, it seems to be the consensus of opinion that unemployment can be reduced and business can be stimulated by public and institutional construction.

The federal government sponsors such a plan. Public opinion has forced construction of schools, town halls, roads, bridges and institutional buildings. Enlightened philanthropists have given money for housing projects, churches and colleges.

Architects, contractors, engineers and business interests profiting from this stimulation of building owe much to the general public. Their responsibility is great. Most assuredly, they ought to eliminate all unfair trade practices and they should observe a high code of ethics. Public favor cannot be maintained in the face of sharp practice, slip-shod methods or faulty specifications.

In the long run, business prosperity depends upon the average prosperity of companies in each competitive field. One unruly, mismanaged price-cutting organization can virtually ruin a whole industry, particularly if such a concern has large credit resources. Correction of abuses in trade practices should come from within an industry and not by pressure from outside.

Too many laws are being enacted as a result of the short-comings of business. It ought not to be necessary in a civilized state to tack on an immense overhead to protect the public against vendors of goods not suitable for use.

How many private homes are uncomfortable in severely cold weather because somebody "short changed" the job? Are public buildings heated and ventilated in the manner expected by the original owners? There are too many "skimped" jobs in heating and ventilating to warrant indifference.

Heating and ventilating specifications have generally left much to be desired. Either they tend to be exclusive, favoring one manufacturer at the expense of another, or they are so wide open that only the vendors of cheap, unreliable products find it profitable to bid.

There is room for some better way to specify equipment. The American Institute of Architects and technical societies might well be enlisted in such a program. Trade organizations and publications should cooperate. The Steel Heating Boiler Institute has made real progress in establishing its code.

The first requirement of a good specification is to define clearly the results desired, for example, maintaining 70° inside temperature when the outside temperature is 0°. If such a clause is taken seriously, 69° will not be acceptable.

A second case might require a fan to deliver 20,000 c.f.m. at 1½ in. static pressure based on standard air and standard test code. This does not mean 20,000 c.f.m. assumed from guesswork tests, nor does it permit the assumption that the pressure, 1½ in., is the total pressure developed by the system when 20,000 c.f.m., are flowing.

The second feature of a good specification is a precise statement of the essential conditions peculiar to the job in hand. These may be shown on plans. They may call for descriptive paragraphs. The type of heating equipment, that is, whether direct radiation, unit heaters, hot air, etc., is for the owner to decide. Similarly, it is not part of a boiler salesman's duties to guess where the stack ought to be.

After clearly showing the desired results and giving a picture of local conditions, it is possible for all prospective bidders to offer equipment selected from their designs most clearly meeting the exact requirements.

The third step in specification writing might well be the exact details of desired apparatus. It does not seem fair to issue invitations to bid, based on ratings taken from some company's data sheets. Suppose an actual job figured out to require a fan delivering 16,500 c.f.m. Further, suppose the favored manufacturer could not meet this desirably, but could offer 18,000 c.f.m. Why specify 18,000 c.f.m.? Perhaps some equally desirable manufacturer has a standard fan rated to deliver 16,600 c.f.m.

However, it is strictly in order to specify minimum

shaft diameters, sizes of bearings, thickness of metal and similar features so as to exclude cheap flimsy equipment. Of course, the inclusion of such details presupposes integrity in their selection. Jokers have no place in an honest specification.

Proper distinction should be made between essential requirements and superfluous details inserted to favor some one or two bidders. There can be no just criticism of specifications that are strict and complete and which call for high-grade proven articles.

No valid objection can be raised if statements be made that so-and-so's valves are acceptable or such-and-such boilers. Such clauses are helpful to contractors because they are informed where apparatus may be purchased.

Cases may arise when only one manufacturer's equipment will do. Space limitation, harmonious business relations, desirable patented features all have their place.

The writer advocates valuing equipment by several standards. Architects and engineers are certainly ethical if they favor well established companies with sound financial ratings. Guarantees made by weak, unreliable concerns have very little actual value except the moral standard of the management. Death or other causes beyond human control may readily remove the only background of the guarantee. A good past history and financial accountability are assets not to be considered lightly.

Tests may be good, bad or indifferent. The method, the local conditions, the engineer in charge, are all important. Engineers and architects drawing up specifications would do well to inform themselves of tests offered as claims for superior performance.

Manufacturing facilities that produce the equipment are important. This has no reference to size. A large plant may be overloaded with run-down, inaccurate machines. A smaller plant may turn out much better products.

It is for this reason that some statements should be made defining the workmanship, design and materials desired. Thus, it is useless to buy expensive alloys if ordinary cast iron will do, but it is folly to make it impossible to buy a more costly material for use where cheap goods cannot stand up.

In most cases, two or more manufacturers are producing equipment sufficiently competitive in character to warrant mention of more than one acceptable manufacturer.

However, some method of evaluation might be considered to establish the relative economic value of various bids. This is important. Trade association codes of necessity must be confined to minimum requirements and methods of rating. High grade manufacturers will not offer products just meeting these minimum standards. They rightly prefer to incorporate in their designs those additional features that make for long service, low maintenance and elimination of annoyances such as noise, dirt and vibration.

It is much beyond the province of one writer or one engineer to set himself up as an authority on any such comprehensive subject as heating and ventilating specifications. Such a program should be undertaken by organized groups of architects, engineers, contractors and manufacturers.

From a careful study and honest survey of the facts, codes might be developed that would free heating and ventilating from many of the weaknesses of today.

Can any high grade engineer doubt the value of the A.S.M.E. boiler code? The writer recalls the boiler business of over twenty years ago. Boilers condemned as unfit for use by one insurance company would be reinsured by some more speculative competitor. Many boilers offered to the trade would be rejected at sight by discriminating purchasers should their purchasing agents have the temerity to buy on price.

It is freely admitted that hazards from the use of poor quality boilers forced attention to higher codes of construction and State laws. Nevertheless, even in the case of products that do not blow up or otherwise jeopardize safety, the public is entitled to honest bids and to some measure of protection.

At no time have heating and ventilating interests been in a better position to elevate the standing of their industry and profession than now. Any serious move to meet the public with whole-hearted cooperation might well speed recovery of business. Confidence is sorely needed now. Assurances that jobs undertaken at this time would be executed carefully, competitively and honestly would bring prompt rewards to everyone worth while in heating and ventilating.

Hub of America in a Coal Mine

IN a strip coal mine, three miles northeast of Linton, Ind., is being erected a marker to indicate the exact center of population as determined by the recent Census Bureau survey. Thus, the hub of the United States is located in a coal mine, an appropriate coincidence, which may be interpreted as symbolic of the fact that coal is the center of our national economic structure.

Despite the persistent threats of hydro-electric, diesel power, natural gas and fuel oil—coal is the focal point of our national existence. On it depend our basic industries, our transportation, our standards of living, our health, our comfort, and our well-being.

Again it is demonstrated that coal, our basic fuel, is consistently in the middle of things.—*Combustion.*

The Principles of Calculation of Low-Temperature Radiant Heating

(Concluded from February Issue)

By A. H. BARKER

Heat Losses from a Human Body

THE conditions produced must be such that the body, as a whole, is kept in a certain degree of comfort, which may be defined as that combination of air and radiant temperature which would procure a heat loss of the desired total amount from a body placed in these conditions. Where the air and radiant temperature are the same, that is called the "effective temperature," which is said to be a correct criterion of comfort.

Physiological research tends to the opinion that heat comfort in a room depends entirely, or almost entirely, on the rate at which heat is abstracted from the body by the surrounding environment, irrespective of the mode of abstraction.

Judging from my own experience, sensations and observations, my personal opinion is that this statement is not correct. I believe that the feeling of comfort depends to a great degree on the way in which the heat is abstracted and that a change in the radiant loss is of much greater effect on the feeling of comfort than an equal change in the convection loss. There are no definite quantitative experimental results with which I am acquainted to decide this matter. For present purposes it is necessary to accept physiological opinion on this point, which needs confirmation by research.

The "effective temperature" is measured by an apparatus known as an eupathoscope.

The first principles of physics show that the total loss from a human body must be equal to the interior heat production, otherwise there would be an undesirable alteration of body temperature.

It is also well known that the rate at which heat is generated in the body varies greatly, according to the state of activity, and that corresponding heat losses are provided for by nature by various thermostatic means, which we need not now discuss. The most important one is by altering the surface temperature of the body to suit the prevailing conditions. This variation, vital though it is to the validity of the calculations in extreme conditions, is another of the variable factors of which we cannot take account for normal conditions, as enough precise information on the subject does not exist, and the consequent complications would render the calculations practically impossible. We shall assume that the surface temperature of the body is always constant. Any thermostatic adjustment of this function and others made by Nature are all to the good. This assumption only leads to material error in extreme cases, as for instance a body kept warm by radiation in freezing weather, or by hot air in an ice cave, as may be judged from consideration of Table 4, and as explained below.

It is evident, therefore, that to be quite appropriate for the comfort of a human body in different states of activity, different degrees of heat loss and different room conditions are called for.

We shall consider the matter according to three grades of activity, in which the total amounts of heat loss, including evaporation, are as follows:

TABLE 3.

AMOUNT OF HEAT LOSS FROM HUMAN BODY IN DIFFERENT CONDITIONS OF ACTIVITY.

Condition	Total loss	Evaporation	Convection	Radiation	Totals (3) & (4)
(1) Absolute rest. (Basal metabolism.)	245	30	90	125	215
(2) Moderate Sedentary Activity	425	92	138	195	333
(3) Moderate hard work...	546	120	178	248	426

It is evident that the total actual heat loss can in any case be subdivided into any desired manner between radiation and convection by adjusting respectively the mean radiant temperature and the air temperature in the room interior. It is the current theory that provided that the same total heat loss is procured, a similar state of comfort will be obtained however the total is subdivided between the two sections.

Dimensions of the Human Body

In investigating this matter it will be necessary to calculate the radiation loss and the convection loss separately. From some careful measurements I have formed the opinion that the amount of heat radiated from an average body will be approximately equal to that from a cylinder at the same temperature as the body—having an effective radiating surface of about 15 sq. ft. The total superficial area of the whole body will be taken as 21 sq. ft., from the whole of which heat will be lost by convection.

Surface Temperature of Body

The average temperature of the body surface is said to be about 75°, including the surface of the clothes and of the skin.

Such measurements as I have been able to make would have led me to a somewhat lower temperature, about 70°, but as most of the work of the physiologists and others has been done on the assumed figure 75°, I have adopted that as a standard.

In fact, the temperature of the body surface varies with the air temperature and with the character of the clothing, and is not constant; for instance, we know by experience that the hands, face and feet become cold in cold weather. This is one of Nature's devices for retaining the body heat in cold surroundings. But it is impossible to take count of this variation. In order to calculate at all we must assume a constant temperature of 75° and see how far this assumption leads us away from what we know by practical experience to be the truth.

Shape of Body

In fact, also, the heat loss by convection varies greatly with the shape of the body, which is both complicated and continuously varying. Every movement a body makes not only alters its shape but also the velocity of the air passing over it. We shall assume that for purposes of calculating the convection an average body is represented by a vertical cylinder 1 ft. diameter and 6 ft. high, having a total surface of 21 sq. ft.

The problem, therefore, becomes reduced to that of so adjusting the mean radiant temperature and the air temperature, respectively, that the total heat loss by radiation and convection added together is in accordance with the figures given in Table 3. That is the problem before us in concrete shape.

If we take Rubner's figures as the standard for ordinary room warming, we are required so to alter the interior conditions that the mean radiant, and the air temperatures, are simultaneously adjusted to such a point that the heat loss by radiation from 15 sq. ft. of surface, at 75°, will be 195 B.t.u. per hr., and from 21 sq. ft. of surface by convection, 138 B.t.u. per hr.

It will be seen from an inspection of the diagrams, that according to these figures the mean radiant temperature surrounding the body should be 60°. and the mean air, 62°.

But the human body is of such adaptability that it is far from necessary that these conditions should be precisely maintained. If we raise the air temperature above 62°, and so reduce the heat loss by convection, we should be able to drop the mean radiant temperature and thus increase by the same amount the heat loss by radiation, and the resulting degree of comfort according to our theory would be the same. Judging by this criterion, we should find the following pairs of values of mean radiant and air temperature would procure the desired rate of heat loss from the body, as given in Table 4 and shown graphically in the curves, Fig. 3, which are calculated for three grades of activity.

Table 4 and Fig. 3 show in each case the combined values of the mean radiant and air temperatures which would procure a uniform total loss of heat of the given amount from a vertical cylinder of 12 in. diameter maintained at 70°, of which the effective radiating surface is 15 sq. ft. and the total surface from which heat is lost by convection is 21 sq. ft., assuming the accuracy of the laws which have been described, and assuming a constant temperature in all conditions. The curves are drawn for three standard rates of heat loss as defined in Table 3.

In an ordinary closed room with more or less regulated ventilation, not exceeding two or three interchanges per hour, the difference between the radiant and air temperature will always be small, probably not exceeding at any time about 10°, that is to say, that the operative part of the curves for a closed room given in Fig. 3 will be confined to the lines enclosed by the circle, Fig. 3.

It is only in the case of a room like an open-air school where cold ventilation is freely introduced that we have a large difference between the air and the radiant temperatures.

TABLE 4.

RADIANT AND AIR TEMPERATURES CORRESPONDING TO EFFECTIVE TEMPERATURE OF 61°, FOR PROCURING TOTAL HEAT LOSS OF 233 B.T.U. PER HR. FROM 15 SQ. FT. RADIATING SURFACE AND 21 SQ. FT. CONVECTIVE SURFACE MAINTAINED AT 75°.

Radiant temp. ° F.	Corresponding air temp. ° F.	B.t.u. radiated loss B.t.u. per hr.	B.t.u. convected loss B.t.u. per hr.	Radiant temp.	Air temp.	B.t.u. radiated loss	B.t.u. lost by convection
30	96	555	— 222	75	44	0	333
35	90	495	— 162	76	43	— 15	348
40	85	443	— 110	78	40	— 43	376
45	79	385	— 50	80	38	— 71	404
50	74	325	8	82	35	— 101	434
52	72	301	32	84	33	— 135	468
54	70	275	58	86	30	— 165	498
56	67	249	84	88	27	— 195	528
58	65	223	110	90	23	— 225	558
60	62	197	136	92	20	— 255	588
62	60	171	162	94	18	— 285	618
64	58	147	186	96	15	— 315	648
66	55	121	212	98	12	— 345	678
68	53	97	236	100	9	— 375	708
70	51	70	263				
72	46	22	311				
74	45	15	318				
75	44	0	333				

Observations on Table 4 and Fig. 3

It appears from practical experience that these figures of Table 4 cannot be applicable to a human body in extreme cases, though they may be, and I believe are, correct for a dummy at constant temperature. In an ice cave the mean radiant temperature would be in the neighborhood of 32°, but it is difficult to believe that it would require air to be introduced into such a cave at 96°, as the table shows, to make us feel normally warm. This discrepancy is due in my view primarily to the error in physiological theory that the degree of comfort is a simple function of the rate of heat loss, and also probably to the fact that the surface body temperature would alter in those conditions, and that there would be some natural thermostatic adjustment of the body which would throw our calculations out in these extreme conditions.

There is only a comparatively small portion of these curves that enter into practical engineering, in relation to the steady state in an enclosed room, because the tendency is for the interior conditions of any room warmed by radiation, or otherwise, to become uniform. In any closed room left to itself unventilated, the ulti-

mate effect will be that the air and the mean radiant temperatures will tend to the same value.

If the heat supplied is mainly by radiation, the air can never exceed the mean radiant temperature. On the other hand, if the heating is mainly by convection, the mean radiant temperature can never exceed that of the air. The amount of the difference depends on the degree of ventilation. If there is none, the excess will ultimately be nil, and if it is limitless the air temperature will become equal to that of the outside air. Any difference is possible between these limits.

Also, the degree by which the radiant exceeds the air temperature depends on the proportion between radiation and convection, characteristic of the particular type of radiant element used.

When radiant surfaces are placed on the ceiling, the convection is very largely suppressed, since warm air will not in the ordinary way circulate downwards from the ceiling to the floor. The proportion here is 80 to 20. In such a case the convection effect is almost entirely secondary. The warm air from a surface so fixed tends to circulate over the whole area of the ceiling, and by increasing the surface temperature there add considerably to the secondary radiation.

This is the explanation why ceiling fixing is extraordinarily economical in fuel. Another reason is that all the heat, of whatever kind, emitted by panels so fixed is ultimately delivered into the center of the building in radiated form. Some of this heat passes by conduction to the floor over, and though it is not used in the room in which the panel itself is fixed, it is nevertheless utilized as low temperature floor radiation in the room over.

Calculation of Mean Radiant Temperature

We must now consider the practical methods of calculating the value of the mean radiant temperature.

For practical purposes, where there is no great variation in the wall temperatures, it is nearly enough the average temperature of the walls. It can be calculated as follows:

Approximate Method. Divide the entire inside surface area of the walls, ceilings and floor into a number of separate portions, each of which is nearly uniform in temperature all over. Multiply the number of square feet in each such portion by its surface temperature in degrees Fahrenheit. Add all these products together and divide the sum by the sum of the surfaces, that is by the whole inside surface area of the room. We thus obtain a figure for the mean or average radiant temperature.

Accurate Method. But this process will not serve where there is a great difference in temperature between different parts of the surface because, as can be seen from the curve, Fig. 1, the amount of heat emitted per square foot by a surface at a high temperature is greater in proportion to the temperature difference the higher the temperature is. It rises as the fourth, and not as the first power of the absolute temperature, so that though at low temperatures the difference is insignificant, a small surface at a high temperature has a much greater proportional effect on the radiant temperature than the same amount of surface at a lower temperature.

In this case of a room warmed by hot radiant elements, therefore, we have to adopt another process of calculation which will be explained below.

Example of Calculation. In order to illustrate the principles of this method of calculation, a drawing of a typical room in a modern school is shown in Fig. 4 with orthodox calculations, and also the details of the calculations which would be made in accordance with principles described in this paper. Let us be quite clear in the first place that what we are calculating is *what amount and temperature of flat radiating surface will raise the room, in an assumed initial cold condition, immediately to a higher condition of radiant temperature when it is turned on in the morning* and not necessarily what is an appropriate amount for practical purposes. The latter consideration depends on a judicious choice of the assumed initial and final temperatures.

The *first process* is to estimate what mean radiant temperature may be expected to prevail in the room after it has been unheated throughout the previous night. The data for this calculation are the initial surface temperatures of all the interior walls of the room, including in that expression both the windows, the ceiling and the floor. The actual temperatures are a matter of considerable uncertainty. They will depend to a large extent on the material of which the walls are constructed, on the number of hours cooling to which the room has been subjected during the previous night, and also both on the external temperatures and on the degree of ventilation which has been allowed during the night.

If the windows of the room have been left open throughout the night, as is often done for purposes of ventilation, the interior surface temperatures may be expected to be considerably lower than these figures. Also, the material of which the wall is constructed, especially the external wall, has a considerable influence on the interior temperature. A variation in these initial interior temperatures will make a corresponding difference in the result of the calculation, and they must be, in the long run, determined by experience.

It would, in fact, be impossible to lay down definitely what these internal temperatures will actually be, as their value depends on many conditions which cannot be known and over which the heating engineer has no control. Experience will show what initial values must be *assumed* in order that the result of the calculation may be an appropriate amount of surface for practical use, for the production of specified conditions in a specified time.

The calculation is here made to demonstrate the principle which must be applied to each problem as experience dictates. It is not intended to lay down here the precise values of the various figures. These must be ultimately determined by research.

As given here it represents the *immediate* result on the radiant temperature of introducing into the room in the assumed condition an amount of radiation given by the calculation. Thus, assuming that the mean radiant temperature is initially at the value calculated below, then so soon as radiating surface of the calculated

amount is heated up to the temperatures specified, the radiant temperature will be immediately raised to the desired value, and will continue to rise above that value. The air temperature will simultaneously begin to rise, partly due to primary convection and to a less extent to secondary convection, until the ultimate value of the effective temperature reaches the steady state. The values so found have to be modified according to the steady state desired and to the time in which it is required to raise the temperature.

The first operation is to calculate from the assumed temperature of the walls the value of the initial radiant temperature. Multiply the area of each of the sections into which the environment is divided by its assumed surface temperature

of the window glass will be uniformly	45°
of the outside walls	50°
of the inside wall	55°
and of the ceiling and floor	55° each.

These total areas are multiplied by their respective temperatures, the resulting products added together, giving 107,000 sq. ft. degrees, the total inside area being 2,016 sq. ft. On dividing the total, 107,000 sq. ft. deg. by 2,016 sq. ft., we obtain the approximate mean radiant temperature 54°.

Now, if we require for purposes of comfort that the mean radiant temperature shall be *immediately* raised to a comfortable degree, we have to calculate how much exposed surface must be introduced into the room, and at what temperature.

This is done as follows, and for the following reasons:

A standard body of 15 sq. ft. at 75° in the middle of the room will interchange with an enclosure at 54° an amount of heat equal to 267 B.t.u. per hr.

According to Rubner's figures the *comfortable* net amount for such a body to radiate is only 195 B.t.u. per hr., which, as has been shown, corresponds to a mean radiant temperature of 60°. Thus, if a body of 15 sq. ft. at 60° were placed in such surroundings, it would neither lose nor gain heat. That is the radiant condition we wish to produce. The amount of high temperature radiating surface which will raise the mean radiant temperature from 54° to 60° is calculated as follows:

Assume that the surface temperature of the radiating elements is 500°, which roughly corresponds to that of a Morganite electric panel element. We see from Fig. 1 that any surface of radiativity 0.9, and at a temperature of 500°, will radiate a gross total of 1,275 B.t.u. per sq. ft. per hr., of 19,125 from 15 sq. ft. Now, a body of 15 sq. ft. maintained at 60° in the middle of the room would radiate, in the same condition, a gross total of 109 B.t.u. per sq. ft. per hr., or 1,635 in all. That is to say, such a body would absorb from a surrounding complete enclosure at 500° (19,125 — 1,635) = 17,490 B.t.u. per hr.

Similarly, when at a temperature of 54° the environment will *absorb* from a body of 15 sq. ft. at 60° an amount of heat equal to (1,648 — 1,578) = 70 B.t.u. per hr.

When the actual surrounding surfaces are partly at 54° and partly at 500° and when the latter are suitably disposed, the net effect on the body may be taken

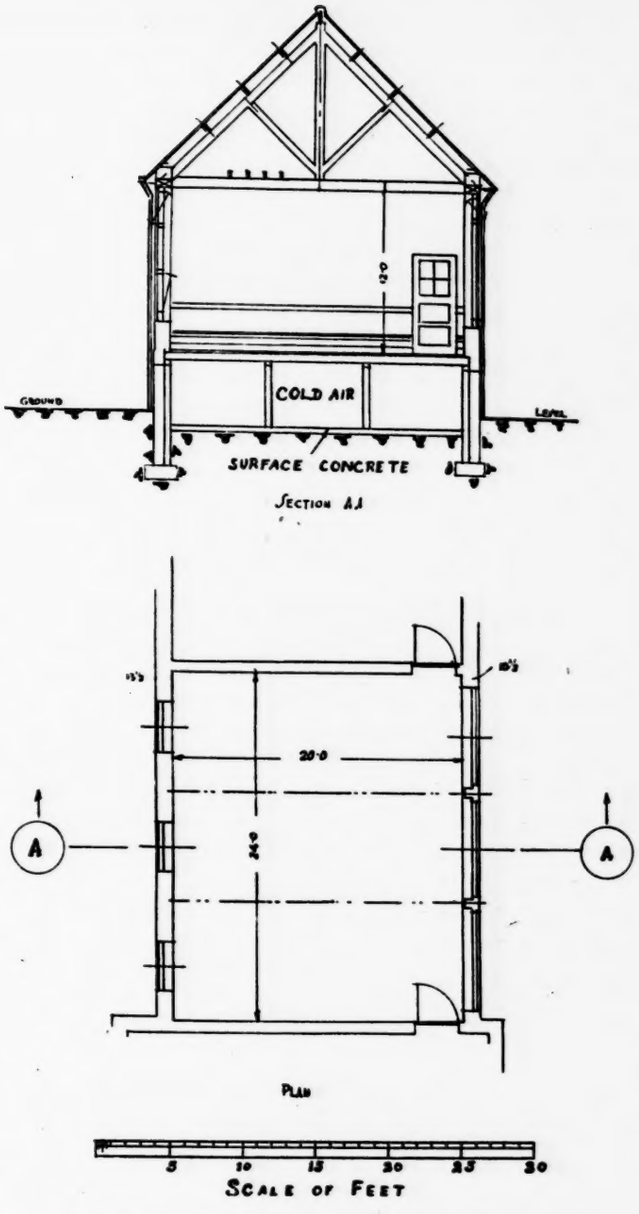


Fig. 4. Typical British schoolroom, calculations for which are given

as jointly proportional to the amount of actual surface at these two temperatures respectively, and to the respective rates at which they interchange their radiation with the body at 60°.

Thus, if a uniform environment, consisting of 2,016 sq. ft., all at 500°, would cause the absorption by the body of an amount of radiation equal to 17,490 B.t.u. per hr., it is evident that a non-uniform environment of x square feet at 500° and 2,016 sq. ft. at 54° would cause the absorption of $\frac{x}{2{,}016}$ multiplied by 17,490, and this quantity of radiated heat is required to make up for the difference of 70 B.t.u. per hr., which the actual environment at 54° produces in excess of the required figure.

Thus we have the equation:

$$\frac{x}{2{,}016} \times 17{,}490 = 70 \text{ whence } x = 8.1$$

This implies that if 8.1 sq. ft. of flat radiating surfaces, maintained at 500°, were suitably disposed in this room, it would have the effect of instantaneously raising the radiant temperature from 54° up to 60°.

In a similar manner we can calculate what will be the amount of ordinary rayrad surface required to produce the same effect. For this purpose we have only to divide by the corresponding figure for a temperature of 140° (which is about the mean temperature of the surface when the water is at 160°), namely, (3,280 — 1,635) = 1,645. We thus find that we should require 85 sq. ft. of exposed rayrad surface as against 8.1 sq. ft. of Morganite surface. This corresponds to 26 sections 30 in. × 16 in.

Applying the same calculation to ordinary plaster panel surface, in which case the surface temperature is of the order of 120°, we find we should require about 154 sq. ft. for the same room.

The following shows the actual calculation:

Calculations of Typical Room Fig. 4

Contents 5,760 cu. ft.	Glass area 279 sq. ft.
External wall 297 sq. ft.	Internal wall 480 sq. ft.
Floor 480 sq. ft.	Ceiling 480 sq. ft.

Initial Mean Radiant Temperature.

Windows	279 × 45 =	12,550
Outer wall	297 × 50 =	14,850
Inner walls	480 × 55 =	26,500
Ceiling	480 × 55 =	26,400
Floor	480 × 55 =	26,400
Total surface	2,016 sq. ft.	106,700 sq. ft. deg.

$$\text{Mean radiant temperature} = \frac{106{,}700}{2{,}016} = 53.6°.$$

Desired mean radiant temp.	60°
Desired air temp.	63°

Loss of heat by radiation from 15 sq. ft. dummy at 60°. (see Table V) 267 — 197 = 70 B.t.u. per hr.

Radiating Surfaces

(1) *Morganite panel.*

Radiation from surface at 500° (Table 1) 1,275 B.t.u. per sq. ft.

Heat absorbed by dummy at 60° from complete environment (2,016 sq. ft.) all at 500°.

$$(1{,}275 \times 15) - (109.5 \times 15) = 19{,}125 - 1{,}635 = 17{,}490 \text{ B.t.u. per hr.}$$

Required heat absorption by dummy 70 B.t.u. per hr. (to balance loss).

If x = required surface:

$$\frac{x}{2{,}016} = \frac{70}{17{,}490} \qquad x = \frac{70 \times 2{,}016}{17{,}490} = 8.1 \text{ sq. ft.}$$

(2) *Electrorad.*—Surface temperature 250°.

$$(381 \times 15) - (109.5 \times 15) = 5{,}750 - 1{,}635 = 4{,}115 \text{ B.t.u. per hr.}$$

$$\frac{x}{2{,}016} = \frac{70}{4{,}115} \qquad x = 34 \text{ sq. ft.}$$

Rayrads.—Surface temperature 140°.

$$(219 \times 15) - (109.5 \times 15) = 3{,}280 - 1{,}635 = 1{,}645 \text{ B.t.u. per hr.}$$

$$\frac{x}{2{,}016} = \frac{70}{1{,}645} \qquad x = 85 \text{ sq. ft. or 26 sections.}$$

Plaster panel heating.—Surface temperature 120°.

$$(169.5 \times 15) - (109.5 \times 15) = 2{,}550 - 1{,}635 = 915 \text{ B.t.u. per hr.}$$

$$\frac{x}{2{,}016} = \frac{70}{915} \qquad x = 154 \text{ sq. ft.}$$

Floor heating.—Surface temperature 70°.

$$(119.5 \times 15) - (109.5 \times 15) = 150 \text{ B.t.u. per hr.}$$

$$\frac{x}{2{,}016} = \frac{70}{150} \qquad x = 930 \text{ sq. ft.}$$

This result is applicable to upper part of room only. Half this quantity would suffice if the radiant temperature is measured near the floor on account of the proximity of the warm surface.

Tubular heating.—Surface temperature 180°.

$$(251 \times 15) - (109.5 \times 15) = 3{,}750 - 1{,}635 = 2{,}115 \text{ B.t.u. per hr.}$$

$$\frac{x}{2{,}016} = \frac{70}{2{,}115} \qquad x = 66 \text{ sq. ft.}$$

Four column radiator.—Surface temperature 160°.

$$\frac{22{,}400}{160} = 140 \text{ sq. ft. (ordinary consecutive calculation)}$$

Emission of Heat by Radiation and Convection from Above Surfaces.

Sq. ft.	Name	Temp. F.	B.t.u. hr. radiation	Convection	Total B.t.u. supplied
8.1	Morganite panel ...	500°	9,500	6,100	14,600
34	Electrorad	250°	9,100	6,100	15,200
85	Rayrad (wall)	140°	9,200	6,150	15,350
154	Plaster panel	120°	9,240	2,310	11,550
930	Floor heating	70°	9,300	4,000	13,300
66	Tubular	180°	9,200	9,200	18,400
140	4 Col. radiators ...	160°	3,320	19,680	22,400

Simplification of Calculation

The above calculations have been made in this extended form in order that the underlying principle may be grasped. If we analyze the matter still further, it will be seen that the amount of radiating surface required at any temperature is, in each case, proportional to the amount of cool surface multiplied by *its negative radiating effect* on a body maintained at the desired mean radiant temperature. Further, that is not necessary to introduce the factor 15, which runs through the whole of the calculations, so that we can determine the amount of radiating surface necessary, independently of a formal calculation of the exact value of the mean radiant temperature or of the gross heat lost from a 15 sq. ft. dummy.

Thus, we can say that 1 sq. ft. of glass at, say 46°, will produce a negative radiating effect on the body at 60°, or 65° (according to the assumed value of the desired mean radiant temperature) in the middle of the room, which is equal to the positive radiating effect from a surface of 0.0134 sq. ft. maintained at 500°.

We can similarly equate the wall surface, the ceiling surface and the floor surface in the same way, to a certain amount of hot surface depending only on its temperature and its radiativity.

Assuming, therefore, that the general radiativity of all these surfaces is 90%, we can set up tables, like Table 5, giving the equivalent amounts of surfaces at different temperatures which are required to compensate for different forms of standard window, wall and ceiling surfaces at lower temperatures. We need not

TABLE 5

RADIATING SURFACE REQUIRED TO RAISE UP TO 65° AND 60° RESPECTIVELY THE RADIANT EFFECT OF 1 SQ. FT. OF WALL SURFACE OF 90% RADIATIVITY AT VARIOUS TEMPERATURES.

Temperature of Radiator Surface	Radiant Temperatures	Temp. of wall surface	60°	58°	56°	54°	52°	50°	48°	45°	40°	35°	30°
		65°	.0039	.0052	.0066	.0081	.0095	.0107	.0118	.0134	.0172	.0205	.0235
500°													
		60°	0	.0014	.0028	.0042	.0057	.0069	.008	.0095	.0133	.0168	.0197
		65°	.0064	.0086	.011	.013	.016	.018	.02	.022	.028	.034	.039
400°													
		60°	0	.00225	.0045	.0069	.0091	.0112	.013	.0156	.0218	.0274	.032
		65°	.012	.016	.02	.024	.029	.032	.036	.041	.052	.062	.072
300°													
		60°	0	.0041	.0082	.0125	.017	.0204	.024	.028	.04	.05	.059
		65°	.017	.023	.029	.035	.042	.047	.052	.059	.075	.090	.103
250°													
		60°	0	.0059	.0081	.018	.024	.029	.034	.041	.057	.072	.084
		65°	.0266	.036	.046	.056	.066	.074	.083	.093	.118	.142	.162
200°													
		60°	0	.086	.018	.028	.038	.046	.054	.064	.089	.11	.13
		65°	.049	.066	.084	.102	.121	.136	.151	.170	.217	.261	.298
150°													
		60°	0	.016	.033	.051	.068	.083	.096	.12	.16	.20	.24
		65°	.081	.11	.139	.169	.2	.225	.250	.28	.36	.43	.495
120°													
		60°	0	.026	.052	.08	.11	.131	.152	.182	.254	.32	.38
		65°	.136	.185	.234	.284	.333	.38	.42	.47	.605	.73	.84
100°													
		60°	0	.042	.084	.13	.175	.21	.245	.292	.41	.51	.60

specify what kind of surface it is, but only its temperature.

Each table, however, is only valid for the particular radiant temperature mentioned (in this case 65° or 60°). If such tables are to be generally useful, much more information about the actual temperatures of the surfaces, and of the desirable value of the mean radiant temperature, must be obtained. A separate table drawn up for each must be prepared if the 60° is found unsuitable. As has been said, it is impossible to know beforehand what these wall temperatures will actually be, and for practical purposes it is unnecessary. But it is necessary at the outset to determine what temperatures *should be assumed* in order that the ultimate result reached after, say two hours' heating, should be satisfactory.

The figures given represent my own entirely independent opinion on this point, derived from a good deal of experience in my own practice, and as will be seen later, they appear to correspond closely with the best current practice derived solely from empirical experience by several leading firms of heating engineers.

It should be observed, in this connection, that there will be a progressive increase in the radiant temperature initially established in a room, if the hot surfaces are maintained hot for a long period. The amount and rate of this progressive increase, though it cannot be exactly calculated, can be estimated with some approach to reasonable accuracy. Thus, in the present case, we see that the amount of heat radiated from the 8.2 sq. ft. of Morganite surface calculated will be of the order of 10,300 B.t.u. per hr.

If this is assumed uniformly distributed over the whole of the interior room surface of 2,016 sq. ft., the average amount available per hour for each, per square foot, will be roughly 5 B.t.u. The exact amount depends on the orientation of the various surfaces.

We know that the rate at which heat is conducted through walls of various types depends on their thickness and on the difference in temperature between inside and outside. For a 14 in. outside wall and a 30° difference, the rate of heat loss is about 8.43 B.t.u. per sq. ft. per hr., the difference in surface temperature between inside and outside being 20.37°.

If then we cause 5 B.t.u. per hr. to be absorbed by the surface of this wall which weighs 135 lb. per cu. ft. and has a water equivalent of 27 lb., it is not unreasonable to expect that the rise in temperature per hour will be generally of the order of 0.4° per hr. The increase in ultimate temperature over that which would be produced by a steady state of interior air temperature will be roughly 4°. Hence, in the steady state we can estimate on ultimately increasing the initially calculated radiant temperature by this amount when the steady state is reached.

These approximate calculations are naturally of a rough character, and they do not involve any considerable application of mathematical physics, but the result cannot be so far in error as to vitiate such calculations as are required for purposes of the engineer.

Air Temperatures

We have now to estimate the effect of these radiators on the air temperature.

Here we are met with the same difficulty as is always met with in heating calculations, that we have only a vague idea of the actual rate of air interchange. Attention has repeatedly been called to the fact that the uncertainty of the rate of air interchange throws all our calculations, of whatever kind, and on whatever principle based, into the region of uncertainty. No man can possibly foretell what the actual rate of air interchange in a room is going to be at any given moment.

It is, therefore, assumed in this case that there will be two interchanges per hour, and on that assumption the ultimate rise in temperature of air, due to primary convection, will be determined as follows:

The proportion between convection and radiation from a flat surface at 500° will be about as 63 is to 37, so that the heat available for convection from the whole of the calculated surface will be 5,100 B.t.u. per hr.

As the interchange of air is 10,000 cu. ft. per hr., this corresponds to an ultimate rise in air temperature, due to primary convection, of 28.5° or 58.5° in all less the heat lost by contact with the walls. But one of the peculiarities of radiant heating is that some of the radiant heat is practically used twice, once to raise the radiant temperature and afterwards to raise the air temperature. Any of such heat which falls on the surfaces of furniture and on that of the walls, not only functions as radiant heat, but raises these surfaces in varying degrees above the air temperature, which in turn warm the surrounding air. The amount of this "secondary convection" can only be roughly estimated. It is totally impossible to calculate it, as it depends not only on the surface of the furniture in the room, but also on its disposition with regard to the radiating surface.

It is, therefore, nothing more than a guess that 20% of the radiant heat emitted will be subsequently utilized as convection, say 1,900 B.t.u. per hr., and this will bring the maximum air temperature up to the value 69° less the heat lost to the cold parts of the walls.

It will thus be seen that in so far as this temperature is above what is calculated as necessary, we shall have that much of unnecessary heating, which would be dealt with in electrical heating by the thermostat, with hot water heating by modification in the boiler temperature, or by turning off some of the rayrads.

This calculation takes no account of the heat lost from the bodies of the school children. In practical life this would, of course, correspond to a further considerable increase, but as the tests would always be made with the room empty, and as the amount of such an increase is very uncertain, it has been left out in this estimate.

Rate of Rise of Temperature

We have also to estimate the rate of rise of effective temperature from such an installation as is under discussion. This, though a matter of great importance, is so complicated that it defies accurate calculation. It is approximately the sum of the rates of rise of the radiant and air temperatures. The immediate rise on the introduction of the radiation is about 6°.

The gradual later rise in the initially produced radiant temperature, caused primarily by the progressive absorption of radiant energy by the furniture and the surface of the walls, may be taken at about ½° per hr.

The rate of rise of air temperature in a room could be calculated mathematically if the precise conditions were known, but such a calculation would be an academic exercise only, and its result would probably have little relation to the practical results.

We know, as a matter of general experience, that the usual rate of rise is such that the full calculated temperature is attained in about three hours, at an ever-decreasing rate of rise; thus, we should roughly estimate in this case an initial rate of rise of about 10° per hr. Adding this to the rate of rise of radiant heat, we could estimate that at the end of an hour, after the heaters were turned on, the effective temperature would have risen by 16½° and at the end of another hour by a further 12°.

In connection with a calculation of this kind, there are so many assumptions that I hesitate to put the results forward as calculations, but only as considered guesses.

The actual temperature of the wall surface would rise gradually from day to day.

In order to compare these figures with current commercial practice, wholly unaffected by my own previously formed views on the matter, I have submitted the drawing of the room to some well-known and highly experienced firms of heating engineers, with a request to furnish me with the scheme which they themselves would put forward if they were invited to tender for a first-class job of this character. The comparisons are as follows:

	Contractors' estimates	Calculated as above	Calculated as Table 5 (60° grade)
For Morganite elements	11.2 sq. ft.	8.1	8.7
For Electrorads	40 sq. ft.	34	37
and for Unity tubes	74 sq. ft.	66	68
Panel surface	135 sq. ft. ceiling panel surf.	154	167
Floor surface	480 sq. ft.	930	—
Rayrad surface	83 sq. ft. exposed surface	85	107
Radiator surface "Ideal Classic"	134	—	—

It was proposed that *Morganite elements* be placed partly in the corners of the rooms and partly in the middle of the side walls, between the windows, obviously with the idea of projecting the axis of the radiation in each case into the center of the room.

The Electrorads were to be fixed on the ceiling, so that the whole of the radiant heat might affect principally the center of the room.

The Unity Tubes were to be fixed around the walls. In the latter case about half of the total radiation from the surface of these tubes would be absorbed by the walls and so to a large extent lost. A good deal of the heat would be in the form of convection, in the proportion probably of 50 to 50.

One contractor would have supplied the panel heating, distributed over the ceiling. For floor heating the pipe coils are fixed underneath precast slabs. The temperature of the floor in these conditions would probably reach 80°.

In all cases it will be observed that calculations made on my assumed basis come out somewhat in excess of the commercial ratings, and would probably be excessive. This indicates that the wall temperatures herein assumed are slightly too low for commercial purposes.

It is only by examining such calculations and comparing them with actual results that these temperatures can be adjusted. This would imply that Table 5 would give more appropriate values with the 60° grade than with 65°. So far as my experience goes, Table 5 might serve provisionally for approximately correct calculations. I have carried out many such calculations and found them satisfactory in my own practice.

Air Conditioning

By WILLIAM HULL STANGLE

Part XVII—Automatic Temperature and Humidity Control

By REALTO E. CHERNE†

IT has been pointed out in previous articles of this series that air conditioning systems are designed to maintain certain conditions of temperature and humidity inside with maximum, or near maximum, conditions outdoors. Consequently, for the major part of the year the equipment will have too much capacity. For this reason, it is necessary to devise means of providing for the operation and control of the plant during periods of partial load. This has necessitated the solution of many exacting problems and has led to the development and application of many control instruments.

The present article will be limited to a description of some of the simple control systems, and an explanation of the mechanical action of some of the instruments and valves used in connection with installations which make use of the principle of bringing all or part of the air to a dew point temperature. Such systems must provide for control of at least two temperatures, namely, the dew point and the room. When outside air below freezing is being used in relatively large quantities, it is necessary also to provide a control which will prevent this cool air doing injury to the equipment. Control of the dew point temperature and the room dry bulb temperature results in a fairly well fixed relative humidity. In some applications, especially in certain textile mills, variations of relative humidity must be carefully limited. In such cases, direct control of the relative humidity is practiced instead of using control of dew point and room air temperature.

In virtually all cases the temperatures to be controlled are some distance from the apparatus which must be adjusted to accomplish the control. For this reason, the temperature-sensitive elements act as pilots and actuate secondary devices which cause power to be applied to the apparatus and accomplish the desired movements to bring about control. The temperature-sensitive elements may be built on any principle which will set up a mechanical movement with temperature change. At least four such principles are in common use commercially, and many ingenious devices have been built. In nearly all cases in air conditioning work the power to move or adjust the control devices is not furnished by the temperature-sensitive element itself and must be supplied. Two sources of power are in common use, viz: compressed air and electricity. While continued improvements in electrical devices have caused them to come into some prominence, by far the greater number of controls used in air conditioning work now utilize compressed air as the motive power. For this reason only compressed air systems will be described in this article.

Before considering any of the several hook-ups of apparatus, it is well to understand the nature and operation of the several pieces of equipment which go to make up a circuit.

† Carrier Engineering Corp.

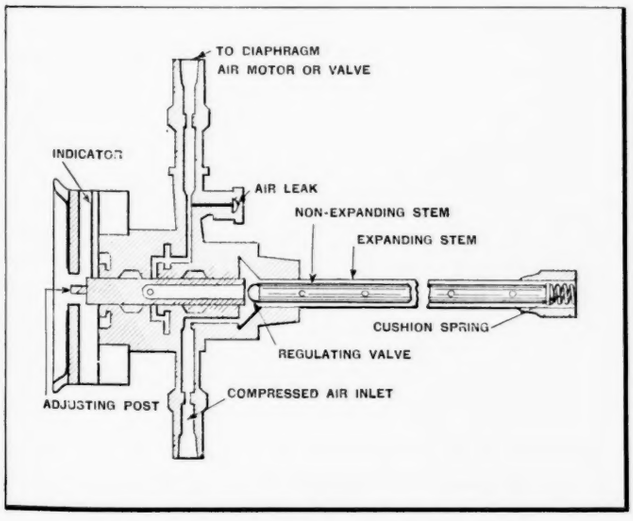


Fig. 1. A graduated thermostat operated by compressed air

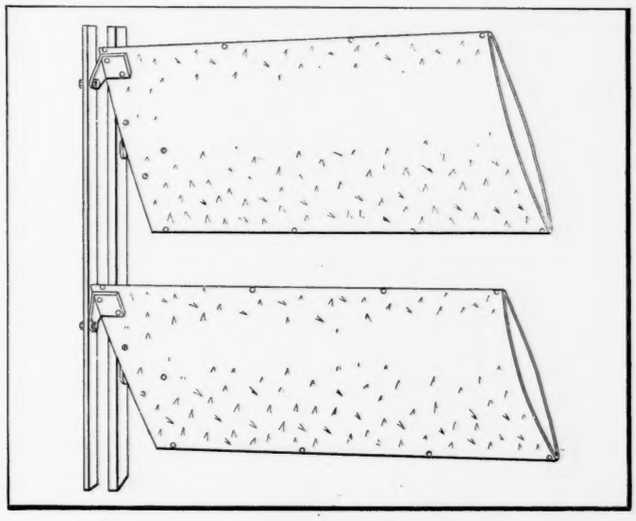


Fig. 2. Louver built in convex style

A simple compressed air temperature control consists essentially of (a) a supply of compressed air, (b) a thermostat or temperature regulator, (c) a compressed air actuated valve to control the flow of steam or water, and (d) an air motor to actuate dampers in the path of air flow.

Every circuit normally contains items (a), (b) and either (c) or (d). Some circuits may contain all four items. Compound circuits can be arranged which introduce many combinations of these elements, but only relatively simple circuits are described here. Some typical forms of commercial apparatus used for each of these items will now be described and illustrated in some cases.

(a) Compressed Air Supply

A small compressor furnishes air at a pressure of 15 lb. to 25 lb. per sq. in. to a receiver or storage tank. The air enters the tank at the bottom through a perforated pipe, bubbling up through water which is contained in the lower part of the compressor tank. By this means the air is cleansed and excess moisture, dirt, oil or scale is removed. The air pressure in the tank is maintained constant by means of an unloader on the compressor and a relief valve on the tank itself. Usually, a pressure reducing valve is located at the tank outlet, so that the air as it passes to the control instruments is expanded, thus avoiding condensation of moisture at the control instruments, and insuring a uniform air pressure throughout the control piping.

From the compressed air supply tank as many separate lines as desired may be run to apparatus. The commonest method of piping is to use a trunk line of compressed air, taking branches from it as necessary in order to reach separate pieces of apparatus.

(b) Thermostat or Temperature-Sensitive Element

The function of this device is to set up a mechanical movement with change in temperature, the movement being used to control the flow of compressed air. The thermostats in use are of two types—graduated-acting and quick-acting.

Graduated-acting: One type of thermostat commonly used is illustrated in Fig. 1 and consists essentially of an outer expanding stem and an inner non-expanding stem. These two members are rigidly connected at one end. The other end of the inner, non-expanding member is provided with a valve, ground to fit an adjustable valve seat. Between the inner and outer tubes is an

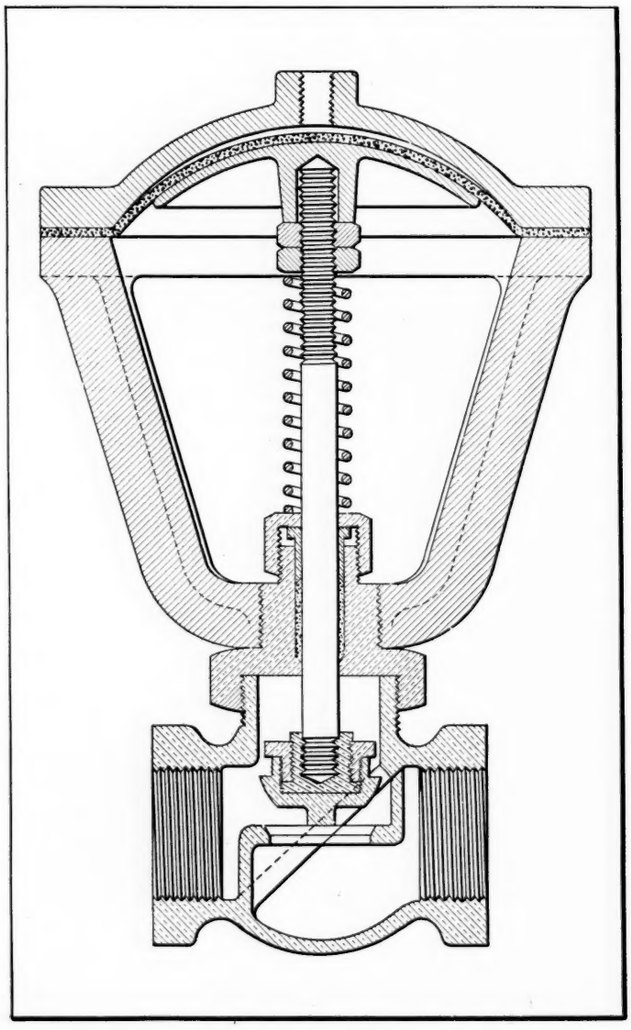


Fig. 3. Compressed air actuated direct acting valve

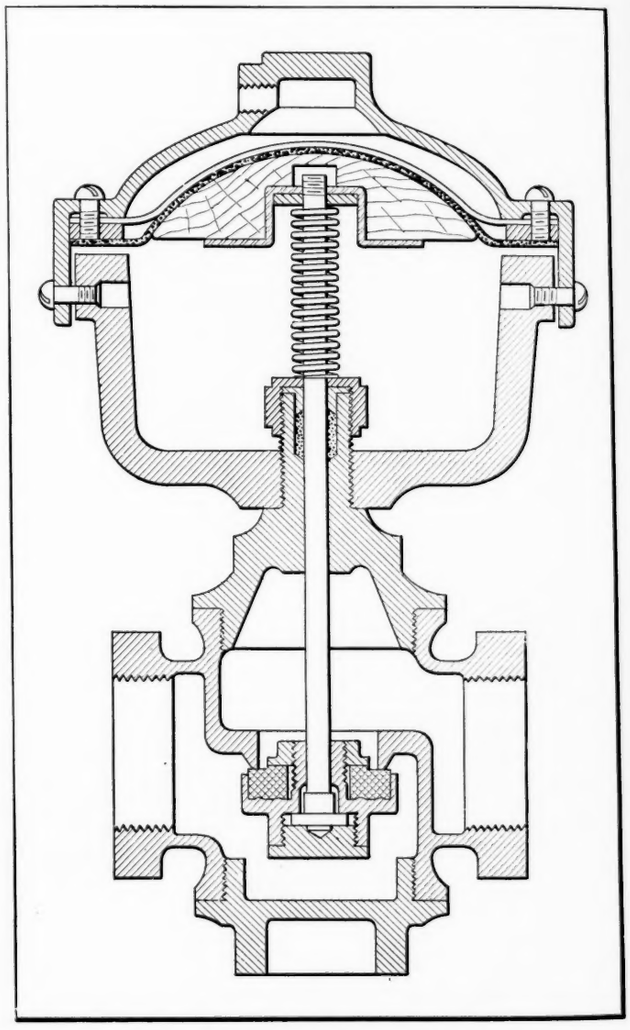


Fig. 4. Three way automatic valve

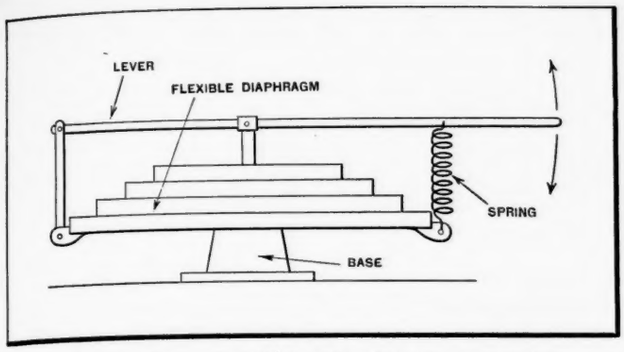


Fig. 5. Compressed air actuated damper motor

annular chamber. Compressed air is admitted to this annular space, and its passage through the instrument is regulated by the small valve attached to the non-expanding stem.

As the temperature of the air surrounding the stem of the instrument rises, the outer member expands, the regulating valve moves from its seat, and the compressed air passes into the outlet chamber, from whence it goes to a valve in the steam or water line or to an air motor actuating a damper in the path of the air flow.

In this thermostat the movement of the valve stem is gradual and the flow of air through the valve is graduated.

Quick-acting: The quick-acting thermostat consists of a device which causes the port opening to the compressed air flow to open fully and suddenly at the temperature at which the device is set. This distinguishes it from the gradual movement of the other type of thermostat. Somewhat the same action can be accomplished by the use of spring-loaded valves.

Both the quick-acting and the graduated-acting thermostats are used in air conditioning control systems. The graduated-acting is particularly useful in the control of the dew point temperature circuit, while the other is used on the circuit which prevents damage when air at freezing temperature is handled through the system.

(c) Compressed Air Actuated Valves

These valves are built in various styles and patterns and are inserted into steam or liquid lines. The compressed air, being at low pressure, must ordinarily act over a considerable area in order to develop enough force to move the valve stem. The air is usually admitted to a chamber one side of which is composed of a movable diaphragm of metal or rubber. The position of the diaphragm depends on the air pressure admitted to the chamber. These valves, therefore, are commonly called diaphragm valves. Fig. 3 illustrates such a valve which is of the globe pattern. These valves are made in the direct-acting and reverse-acting styles.

Direct-acting: In this valve, as shown in Fig. 3, the stem is provided with a curved cap and supported in an open position by means of a spiral spring. Admission of compressed air, through the opening in the top of the valve, causes the diaphragm to seat the valve. When the air pressure upon the diaphragm is released, the spring opens the valve. To insure quick opening, the valve is so installed in the line that the steam or liquid enters beneath the valve. Hence, the pressure of the fluid assists the spring to open the valve. The air pressure is thus working against both the spring and the fluid pressure. By utilizing valve tops of proper size, the closing effort of the air pressure is balanced against the opening effort of the spring and the fluid pressure, so that the valve may take any intermediate position between fully opened and fully closed, depending upon the air pressure admitted by the thermostat.

Reverse-acting: The construction of the reverse-acting diaphragm valve is similar to that of a direct-acting with the exception that the valve seat must be inserted from the bottom of the valve casing. Both direct-acting and reverse-acting relays may be obtained.

Reverse-acting valves are useful in cases where temperature must not be allowed to rise above predetermined levels, or where accidental pressure failures must be guarded against.

Thus, to prevent over-humidifying or over-heating the building in case of possible failure of air or water pressure, a small double-seated safety air valve operated by a diaphragm and coil spring may be put into the compressed air line and its diaphragm subjected to water pressure from the circulating pump discharge. Should the water pump be shut down, or for any reason lose its suction, the air pressure is shut off, which closes a reverse-acting steam valve and shuts off the steam supply. Likewise, should the air pressure fail the reverse-acting diaphragm valve is closed.

Three-way Automatic: An interesting compressed air actuated valve is the one embodying the three-way feature. Fig. 4 shows one form of such a valve. Use of this valve permits rather careful control of temperature of a fluid by mixing as it passes through the valve.

Suppose that a cold water supply is connected to the right-hand opening, and a supply of warmer water to the bottom opening. The water leaving the left-hand opening can have its temperature controlled intermediate to the temperature of the cold and warm water, by the action of a thermostat connected to the compressed air chamber at the top.

Thus, if the temperature of the air at the thermostat is below that desired, the thermostat closes and relieves the pressure upon the diaphragm of the valve. The spring lifts the stem and the upper valve closes the

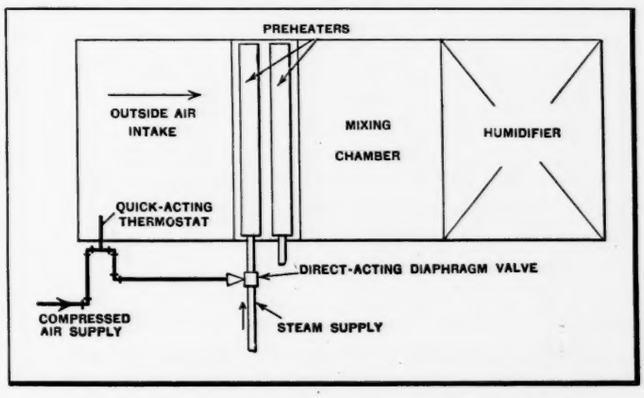


Fig. 6. Layout of circuit for controlling outside air temperature

cold water, leaving the warmer water open to flow through the left-hand opening. If the temperature of the air rises above that desired, the thermostat opens the compressed air to the diaphragm and forces the valve stem downward, closing the warmer water and leaving cold water open to flow through the left-hand opening. Obviously, this type of valve is readily adapted to the gradual action necessary for uniform regulation of the temperature of the fluid flowing out of the left-hand opening.

As an example of the use of this type of valve, suppose that control of the apparatus dew point is sought at a time when refrigerated cold water is necessary to dehumidify the air passing through the conditioner. The problem resolves itself into accurately maintaining the temperature of the cooled water which is pumped to the spray headers in the conditioner.

A thermostat is placed in the path of the air leaving the conditioner and is adjusted to maintain the desired dew point temperature.

The compressed air line is led from the thermostat to the top of the three-way valve. The cooled water supply is connected to the right-hand opening. The bottom opening is connected to the collecting tank in the bottom of the conditioner. The left-hand opening is connected to the suction side of the pump which supplies the spray headers. As the water from the bottom of the conditioner is a little warmer than the cold water supply, which is itself held near the desired temperature, the action of the three-way valve permits the mixing of the water and results in water flowing to the pump at the desired temperature subject to the action of the thermostat.

(d) Air Motors and Dampers

In order to adjust the amount of outside air, return air, by-pass air, or supply air, it is necessary to provide suitable dampers. The dampers may be adjusted or moved either manually or automatically. Where automatic control with compressed air as the motive power is used, the dampers are connected through mechanical linkages to a motor arm which is moved by the compressed air subject to the action of a thermostat which admits air to the motor in the same way as to the valves just described.

Dampers for use in air passages may be either of the single-leaf type or of the louver type. If of the louver type, each leaf must be connected by linkage to the air-motor arm. The louvers may be constructed either of flat pieces or they may be so formed that the surfaces are convex.

Fig. 2 illustrates a louver built in the convex style. This method of construction results in a strong rigid damper, which may be readily mounted on ball bearings. When so built these louvers have the advantage of operating with little power and are free from chattering or generating noises.

Compressed air actuated motors for moving dampers automatically are built in a variety of styles. Fig. 5 illustrates diagrammatically the principle of such motors. Air is admitted through the controlling thermostat to the flexible diaphragm of the motor. The air pressure expands the diaphragm and moves the free end of the arm upward. The damper louvers are connected mechanically to this arm and move as it moves. When the air pressure is relieved, the spring tends to draw the motor arm down. The position of the motor arm thus depends on the air pressure in the diaphragm. The necessary force to cause proper movement is secured by altering the size of the diaphragm and the length of the motor arm. These motors may be secured with diaphragms made of either metal or rubber, as desired. In some applications rubber has but short life because of the conditions to which it is exposed. In such cases the use of metal diaphragms is advantageous.

These motors are obtainable in several stock sizes. Where the dampers to be moved are so large that the power is beyond the capacity of a single motor, more than one motor must be used. In such cases. the motors may be arranged to operate simultaneously by the use of levers and linkages.

Points of Control

It has been mentioned previously that in comfort installations three control points are desirable, viz:

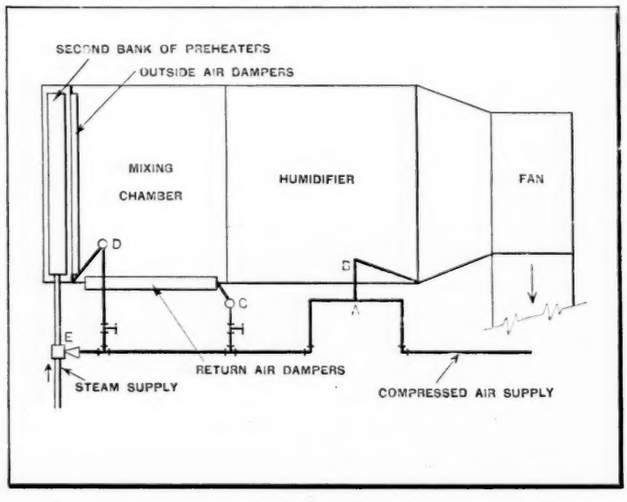


Fig. 7. Method of control during winter operation

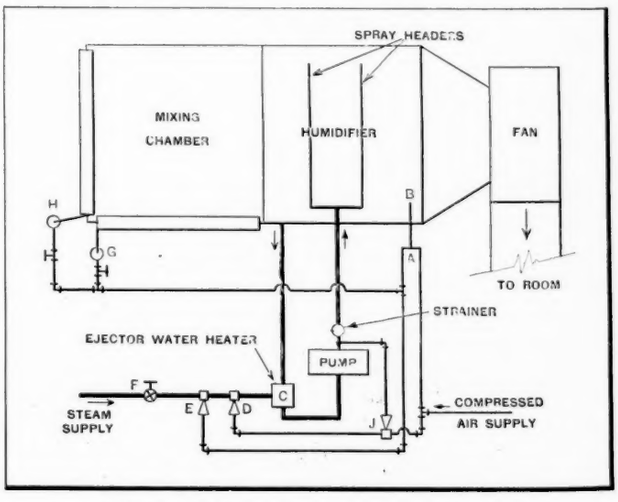


Fig. 8. Ejector water heater dew point temperature control

(a) control of the temperature of the entering supply air when large quantities are handled at low temperature, (b) the dew point temperature, and (c) the room temperature.

With the apparatus elements for obtaining control explained, we will now turn attention to elementary hook-ups of apparatus which may be used for accomplishing the control.

Outside Air Temperature Control

Fig. 6 is a diagrammatic layout of the circuit for controlling the outside air temperature.

In winter the air frequently enters the outside air intake at temperatures well below the freezing point of water (32° F.). A previous article in this series[1] pointed out that the amount of preheating required is dependent upon the relative amounts of outside air and return air used. Ordinarily, two banks of preheaters are required. The outer bank is controlled by a quick-acting thermostat which has a sensitive bulb located in the outside air intake. The setting of this thermostat is such that any time the outside entering air is below a previously determined low point (usually 36° to 40°), a direct-acting diaphragm valve is actuated to allow steam to enter the preheater. The action of this thermostat is quick, that is, the steam is fully turned on or fully turned off at a temperature corresponding to the setting on the instrument. The reason for this is obvious, because any throttling action on the steam control might allow the quantity of steam to drop off to a point where freezing would result in the heater tubes. When the temperature of the entering air goes below the instrument setting, compressed air is allowed to pass through the valve on the thermostat and actuate the direct-acting steam valve. Conversely, when the temperature rises above the temperature set, the air pressure on the diaphragm valve is released and the steam supply is automatically shut off.

Dew Point Temperature Control

Control of the apparatus dew point is complicated by the necessity of controlling it both during times when humidification is required and when dehumidification is essential. Control must be secured over both of these operations.

As pointed out previously, when the saturation temperature or dew point of the air is fixed, its moisture content or absolute humidity is fixed at a value which corresponds to the dew point temperature.

Air which enters the conditioner containing less than the required moisture content (as in winter), must be humidified. Air which contains more moisture than required (as in summer), must be dehumidified. Since the spray water itself acts both as a conveyor of heat to or from the air and as a source of moisture, one method of control is to regulate the temperature of the spray water itself. It is also evident that although the spray water is the medium which directly effects the control, it is the temperature of the leaving air itself which must be controlled. Therefore, the thermostat

[1] Heating and Ventilating, July, 1931.

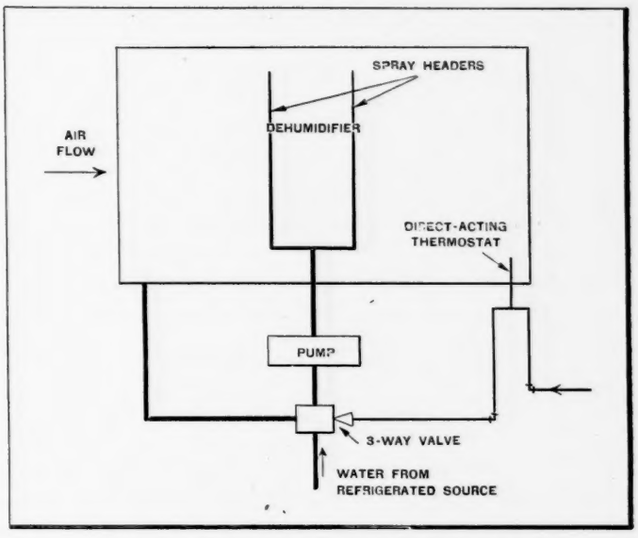


Fig. 9. Method of dehumidifying control

is ordinarily placed in the path of the saturated air leaving the conditioner, and is directly actuated by the temperature of the leaving air.

Winter Control Hook-ups: There are several methods of connecting apparatus for securing control during winter operation.

Fig. 7 illustrates one method that is commonly used in comfort installations. The sensitive bulb B, of the thermostat A is located in the path of the saturated air. When the temperature goes above that desired, compressed air is allowed to pass through the thermostat A, to the direct-acting diaphragm valve E, which cuts off the steam entering the second bank of preheaters to lower the temperature as required.

At times, particularly during intermediate seasons, adjustment of the relative quantities of outside air and return air will give the required dew point without the use of steam or refrigerated water. For example, it may be necessary to use 100% outside air through the conditioner to obtain the required dew point without the use of steam or refrigeration. Should the temperature of the outside air gradually lower, the return dampers gradually open and the outside air dampers gradually close, being actuated by the air motors C and D. At all times an equilibrium is thereby maintained, resulting in maintaining the desired dew point. This action is allowed to continue until the minimum allowable outside air is being admitted, after which steam must be used to supply the required heat.

The opposite condition, that of rising temperature, is handled in a similar manner until the minimum amount of outside air is being used and then refrigeration is required in order to maintain the proper dew point temperature.

There are in common use two other types of winter dew point control in which the temperature of the spray water is controlled directly by the use of steam. One makes use of an ejector water heater which allows the steam to pass directly into the spray water on the suction side of the spray pump, while the other uses a closed water heater in which all or part of the recirculated spray water passes through the heater and receives heat indirectly from steam or hot water.

Of the two methods, the ejector type is the simpler and more compact, but for steam pressure lower than 3 lb. gauge the closed system should be used.

Fig. 8 shows diagrammatically the water circulating system and ejector water heater dew point temperature control. Action of this control is as follows: Thermostat A is placed in the air passage just beyond the eliminator plates and is connected by compressed air lines from the compressed air storage tank to the direct-acting diaphragm valve E, and the outside air and return air damper motors, H and G, respectively. The reverse-acting diaphragm valve D is normally closed, and is placed in the steam line as a protection against over-humidification in case the water or air pressure should fail. The safety relay J is normally open when the water sprays are in operation. Should the water pressure or air pressure fail, the reverse-acting valve would immediately close off the steam supply. Fig. 8 also shows the spray water recirculating system. Water from the settling tank in the bottom of the conditioner is mixed with steam in the ejector heater C, before being drawn into the pump which forces it through the spray nozzles at the required pressure. A hand valve, F, is placed in the steam line to insure a positive shut off of steam during cooling or inoperative periods.

Action of the outside air and return air dampers for this control system is the same as outlined for the dew point control using the preheaters.

When the temperature of the saturated air rises above the setting on the thermostat, compressed air is allowed to pass to the direct-acting valve which causes the valve to cut down on steam. When the temperature goes below that required, air pressure on the automatic valve is reduced and steam is admitted to the ejector heater.

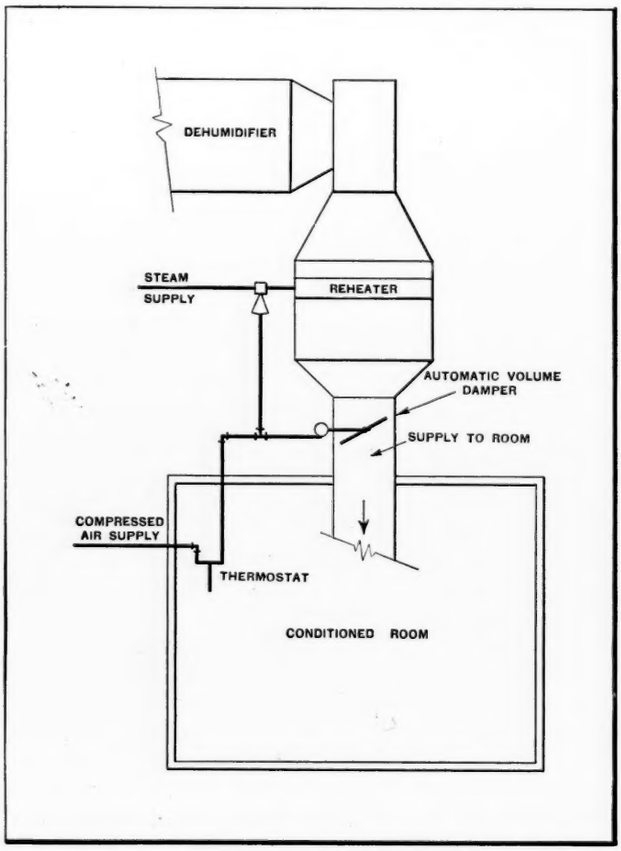


Fig. 10. Control actuated by room temperature

This discussion has been based on using a direct-acting thermostat in the dew point chamber, because this type of thermostat is ordinarily used for summer operation. If a reverse-acting thermostat is used instead of the direct-acting, the direct-acting steam control valve may be eliminated. Thus the thermostat could be connected to the reverse-acting diaphragm valve D, which is normally closed. It opens when air pressure is admitted to it and causes steam to enter the water heater. When the thermostat cools it opens a valve to permit a greater flow of compressed air and, consequently, a greater pressure in the line to the steam valve. In this hook-up the safety relay would have the same action as explained.

The closed water heater control is similar to that described for the ejector water heater except that the compressed air passing through the thermostat operates a three-way mixing valve regulating the flow of water by-passed around the water heater.

Dehumidification Hook-ups: For dehumidification the control is essentially the same as for humidification except that the temperature of the spray water must be decreased rather than increased. This can be accomplished by controlling the temperature of the cold water leaving the refrigeration machine, which can be done automatically by regulating the rate of flow of condenser water or by varying the speed of the compressor. However, in many installations such a method is not feasible due to the operating characteristics of the refrigerating machines and the location of the conditioners. In this event, a three-way valve may be located in the spray pump suction to allow mixing of water from the conditioner settling tank with colder water from the refrigeration source. This control is illustrated in Fig. 9. When the temperature of the spray water rises above the setting of the thermostat, compressed air is allowed to flow to the three-way valve which opens the cold water port and closes off the amount of water being recirculated. Provision must be made to allow the excess water, equal in amount to the refrigerated water being used, to return to the refrigeration machine to be recooled before again entering the circuit.

Room Temperature Controls

A second thermostat in addition to the one placed at the conditioner for dew point control is used to control the temperature of the room. This thermostat is located either in the room itself or in the return air duct, the latter location being used only when a representative sample of the return air is sure to pass over the bulb. When the room and dew point temperatures are thus held constant, the percentage of relative humidity in the room will be maintained constant and at a value corresponding to the difference between the two temperatures.

For comfort installations the room control usually consists of a graduated-acting thermostat actuating volume or by-pass dampers, or a diaphragm valve on the air reheater.

A simple type of room temperature control is illustrated in Fig. 10. During summer operation if the room temperature rises, the direct-acting thermostat located in the room causes compressed air to open the volume damper in the supply duct to the room, thereby allowing a larger quantity of cool air to be admitted to the room. During winter operation, on the other hand, should the temperature rise above that desired, compressed air is delivered to the direct-acting diaphragm valve, causing the steam to be shut off.

Use of the by-pass arrangement at the conditioner permits the quantity of air being supplied to the room to be kept constant. The room thermostat in such a system controls the reheater which is normally located in the by-pass chamber and also volume dampers which regulate the relative amounts of dehumidified and by-passed air. The fan used should be of a type that will not overload itself.

Sometimes it is advisable to use a regulating device to maintain automatically the static pressure in the fan discharge at a constant value regardless of the automatic action in opening and closing dampers. The action of the regulator is such that a change of static pressure within the duct will cause a relay to actuate dampers to return the pressure to the proper level. Ordinarily, the regulator is connected to the dehumidified air dampers and exhaust air dampers or to the by-pass air dampers.

It should be remembered that all room controls must be used in conjunction with proper dew point controls.

Controls for Direct Radiation

In some installations the heat losses are provided for by the use of direct radiators. Automatic control of the heat supply is accomplished by means of the thermostatic apparatus usually used in this connection. In general, these controls are entirely separate from those installed for controlling the air conditioning apparatus, although they may have parts in common.

General Control Design

On large installations, the design of automatic controls is a complex problem. Generally, however, use is made of the instruments and mechanisms described in this article. Any number of hook-ups may be made utilizing compressed air or a combination of compressed air and electrical control. In fact, almost any temperature, humidity, or changing temperature and humidity may be automatically controlled by proper application of thermostats, hygrostats, pressure regulators, diaphragm valves, damper motors and safety relays.

Operating Instruments

In order to facilitate the balancing of a system and also to know the conditions actually existing, it is advisable to install indicating thermometers to register the apparatus dew point temperature, room dry bulb temperature and the dry bulb temperature of the outside air. With these three temperature readings it is possible for the operator to determine whether or not the system is operating properly and if it is not, what adjustments are necessary to bring about a proper condition. The instruments usually used for this purpose are self-contained mercury thermometers.

Some designers prefer to centralize all the indicating thermometers on a gauge board. For this purpose dial type vapor tension or gas filled thermometers are generally used. The action of both types is fundamentally the same. A sensitive bulb partly filled with a volatile liquid or a gas is placed in the medium whose temperature is to be measured. This bulb is connected to the actuating mechanism by means of a capillary tube. When the temperature of the bulb changes, the vapor pressure or gas pressure changes correspondingly. Such changes are transmitted through the capillary tube to the actuating mechanism which registers the temperature at the bulb. These instruments when properly calibrated give accurate temperature readings.

For the purpose of recording the operation, various types of recording instruments may be utilized.

Ewing Galloway

Your Everyday Problems

Mr. Hanburger will be glad to answer your problems relating to installation and operation of heating and ventilating systems, and wherever possible will give the reasons for his suggestions. Letters will be answered direct to the inquirer and in some cases published anonymously. If sketches are included they should be clearly drawn. Write Mr. Hanburger, HEATING AND VENTILATING, 148 Lafayette Street, New York.

Conducted by FRED W. HANBURGER

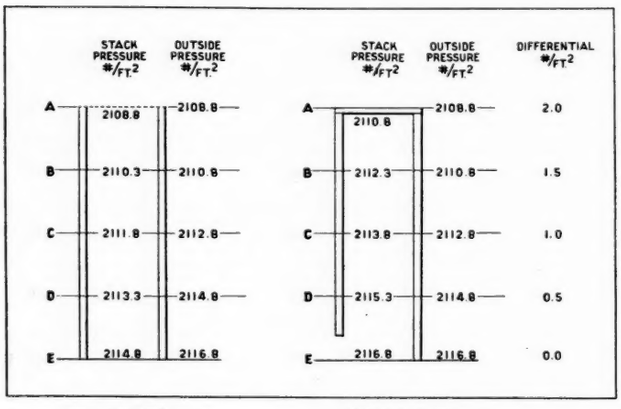
I regret that, due to lack of space, it will be impossible to publish all the letters received on the chimney flow problem discussed here last month. Since no agreement seems to have been reached among the opinions of the correspondents, I have been asked to give my own views on the matter.

The original question was, that as the pressure of the lighter air in a chimney is less than that of the air outside, a hole pierced in the chimney walls at any point should cause the outside air to flow inward. If such is the case and cold air would flow inward through a hole at an infinitesimal distance from the top, is it possible that air will also flow into the top and down the chimney?

Throughout the following discussion a stack 12 in. x 12 in. x 100 ft. high, an outside temperature of 35°, having a density of 0.08 lb., and a stack temperature of 200° F. with a density of 0.06 lb. per cu. ft., will be assumed.

Fig. 1 shows a chimney open at the top and closed at the bottom. The outside air pressure at the base of the chimney is 14.7 lb. per sq. in., or 2116.8 lb. per sq. ft. As the density of the air is 0.08 lb. per cu. ft., the pressure at the top of the chimney will be 100 x 0.08 = 8.00 lb. per sq. ft. less than that at the foot, or 2116.8 — 8.0 = 2108.8 lb. per sq. ft. Corresponding pressures at the mid and quarter sections, points B, C, and D, will be those shown in Fig. 1.

Pressures inside the stack will be less than those outside. The pressure at the top is 2108.8 lb. per sq. ft., but as the density of the air in the chimney is but 0.06 lb. per cu. ft., the pressure inside at the bottom will be 2114.8 lb. per sq. ft., or 2 lb. less than the outside pressure. Corresponding pressures on the inside of the stack, are also shown at points B, C and D. It will be noticed that the difference in the two pressures increases as we move downward, and that a hole pierced at any point of the stack would allow air to flow inward until equilibrium is established.



(Left) Fig. 1. (Right) Fig. 2.

Let us now assume that the top is closed and imagine the walls of the chimney to be of a material having no weight. We have a buoyant effect, the intensity of which is equal to the difference of the weight of the lighter air within the chimney, and the air in which it is immersed, which is 2 lb. per sq. ft. In other words, the stack pressure at A becomes 2 lb. per sq. ft. greater than the outside atmospheric pressure. This is due to the pressure of the outside air on the bottom of the balloon, and is exactly the same condition that would prevail by having the stack closed at the top and open at the bottom. By opening the bottom we allow an added pressure of 2 lb. per sq. ft. to enter, which increases all the internal pressures of Fig. 1 by the same amount. The increased internal pressures are so shown in Fig. 2. It will now be noticed that we have an excess pressure of 0.5 lb. at D, 1.0 lb. at C, 1.5 lb. at B, and 2 lb. at A. The excess pressure is now increasing upward, and a hole pierced at any point in the walls of the chimney will cause the air to flow outward.

We can now omit the absolute value of the pressures, and using only the differentials refer to Fig. 3 to determine the changes that occur when an opening is made in the top and flow established. These pressure differentials are the available heads at the various points, which may be converted into velocity, friction, or lost heads. It must be noted, however, that the problem differs from that of a blower fan where the entire force is generated at one point. In the case of the chimney the force increases as we ascend and the problem is analogous to a downfeed gravity water riser where the head increases as the outlet is approached. At G we will assume a loss of 15 times the velocity head in the stack.

If we now open the stack at the top and the same inside temperature is maintained, circulation is set up, the velocity of which will increase until all the available forces have been used in creating velocity or overcoming friction. Using 0.00000002 as the friction coefficient for a velocity of 1 ft. per min. (which is a good average for square chimneys) it may be found that the resultant constant velocity will approximate 403 ft. per min. The velocity head for 403 ft. per min., expressed in pounds pressure per square foot, is 0.04 lb. Therefore, all the stack pressures must be reduced by this amount, as shown in column 3. If the entrance head at

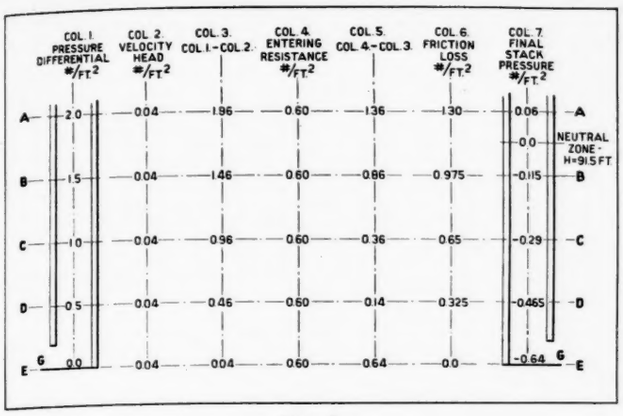

COL. 1. PRESSURE DIFFERENTIAL #/FT.²
COL. 2. VELOCITY HEAD #/FT.²
COL. 3. COL. 1 - COL. 2.
COL. 4. ENTERING RESISTANCE #/FT.²
COL. 5. COL. 4 - COL. 3.
COL. 6. FRICTION LOSS #/FT.²
COL. 7. FINAL STACK PRESSURE #/FT.²
A 2.0 .004 1.96 .60 1.36 1.30 .06 A
.0 NEUTRAL ZONE - H=91.5 FT
B 1.5 .004 1.46 .60 .86 .975 .115 B
C 1.0 .004 .96 .60 .36 .65 .29 C
D .5 .004 .46 .60 .14 .325 .465 D
E G .0 .004 .004 .60 .64 .0 .64 G E

Fig. 3.

G is 15 times the velocity head, then the loss due to entrance will be

$$0.04 \times 15 = 0.60 \text{ lb. per sq. ft.}$$

This again must be deducted from the remaining head, and the results are shown in column 5. There is also an outlet resistance at the top of the chimney which we may assume as 1.5 times the velocity head and, therefore, equal to

$$1.5 \times 0.04 = 0.06 \text{ lb. per sq. ft.}$$

Consequently, the pressure available to overcome friction throughout the entire legnth of the chimney will be 1.36 (at point A) — 0.06 = 1.30 lb. per sq. ft., thus leaving a remaining pressure of 0.06 lb. per sq. ft. at A to overcome the outlet resistance. Now checking the friction by the general formula

$$P = \frac{\text{Perimeter} \times L \times F \times V^2}{A}$$

we have

$$P = \frac{4 \times 100 \times 0.00000002 \times 403^2}{1}$$
$$= 1.30 \text{ lb. per sq. ft.}$$

With a pressure loss of 1.30 lb. throughout 100 ft. of length, the loss per lineal foot will be 0.013 lb., and for 25 ft. of length the loss will be 0.325 lb. This is the friction loss at D, and must be deducted from the remaining pressure of column 5. The loss to C will be

$$50 \times 0.013 = 0.65 \text{ lb.}$$

and at B it is

$$75 \times 0.013 = 0.975 \text{ lb.}$$

while at A it is 1.30. These friction losses are shown in column 6, and the final results after constant flow is established are shown in column 7.

It will be noted that we have a positive pressure of 0.06 lb. at A, and a negative pressure of 0.115 lb. at B, so that the neutral zone lies somewhere between these two points. Its exact location may be determined by noting that the pressure is increasing upward at the rate of 0.02 lb., and is decreasing, due to friction, at the rate of 0.013 lb. per sq. ft. per lin. ft., so that we may write the formula

$$0.02H - (E + V.H.) - 0.013H = 0$$

where E = entrance head, and V. H. = velocity head. Substituting their numerical values and equating, we have 0.007H = 0.64 lb. per sq. ft.

or

$$H = 91.5 \text{ ft.}$$

At this point the air would neither flow in nor out through the hole in the chimney walls.

Let us now assume that a hood has been placed on the chimney which produces a resistance also equal to 15 times the velocity head, with a similar resistance at grate G. The additional resistance lowers the efficiency of the chimney, and it will be found that the velocity will reduce to 350 ft. per min., having a velocity head of 0.032 lb. per sq. ft., giving an entrance resistance of 0.48 lb., and an outlet resistance of a similar amount. By calculation we find that the neutral point lies between B and C, and we further find that the neutral zone is at a height of 51.6 ft.

The velocities of 403 ft. and 350 ft. per min. in the preceding examples were determined by trial, and in all cases must be so determined that the sum of the velocity, friction and all resistance heads will equal the available force. It will be of advantage to note that, as the stack is closed more and more at the top, the air will flow out of the diminished opening with a greater velocity, demanding a greater outlet pressure which is supplied by a diminishing velocity in the stack with less friction loss. The internal pressure then increases, the neutral point falls lower and lower until, when the stack is again entirely closed at the top, fumes would issue through an opening in the walls at all points.

QUESTION 37. I have a hydraulic ram supplying water to my house and am wondering if it would be practical to run the water through the piping of my hot water heating system during the summer months. I have an adequate supply of water. — *M. F.*

ANSWER. If the water is taken directly from a spring it will have a temperature of about 50° and its cooling effect would be quite noticeable. It is not advisable to do this, however, owing to the large amount of condensation accumulating on the radiators. To arrive at an idea of the amount of such condensation let us consider a room 12 ft. x 12 ft. x 10 ft. high, the cubical content being 1440 cu. ft. Further, let us assume a summer day with a temperature of 80° and a humidity of 80%, a condition which often prevails. Air with a temperature of 80° will contain 11 grains of miosture when fully saturated and will have 8.8 grains when 80% saturated. If the temperature of the air passing over the radiator should be lowered to 60°, the dew point of which is 5.8 grains per cu. ft., then each cubic foot of air passing over the radiators will give up, or deposit

$$8.8 - 5.8 = 3.0 \text{ grains per cu. ft.}$$

Allowing one air change per hour for the room, which is low with the open doors of summertime, there will be

1440 cu. ft. x 3.0 grains per cu. ft.=4320 grains, or

$$\frac{4320}{7000} = 0.62 \text{ lb.}$$

of water collected per hour. You will note that this amounts to approximately 1 gal. of water in 12 hr. which must be disposed of.

A better plan would be to have special coils for this purpose in the attic where provision can be made to take care of all the condensation at one place.

News of the Month

Short Course in Air Conditioning at Michigan State

A three-day course, March 22-24, covering the engineering essentials involved in air conditioning and forced-air heating, has been announced by the Michigan State College of Agriculture and Applied Science, East Lansing, Mich.

It is the intention of the sponsors of the course that the discussions and instruction shall be led by men skilled in the practical, everyday applications of the engineering principles involved. A fee of $5 covers the entire expense of registration, enrollment and participation in all classes and entertainment features. All those engaged in warm-air heating work are eligible for enrollment. Those expecting to attend should forward promptly their request for enrollment, together with payment of the fee, to Prof. Lorin G. Miller, Michigan State College, East Lansing, Mich.

Lectures and discussion on the following topics are listed on the tentative program: Importance of Analyzing Building Structure; Use of Coefficients or Factors, and How to Find Them; Infiltration, and B.t.u. Methods of Calculation; Duct Design and Layout; Air Flow; Air Conditioning; The Problem of Design and Layout of a Forced-Air Plant by Dealers.

Discovers New Method for Detecting Gas

Dr. William Crocker, director of the Boyce Thompson Institute for Plant Research, Yonkers, N. Y., has discovered a new method for detecting the presence of manufactured gas in the air in a greenhouse. In cold weather especially, when the surface of the ground is frozen, gas escaping from a leak often travels considerable distances under ground, and when it enters a greenhouse, even in quantities undiscernible to the human nose, may do great damage. Sometimes canary birds are put in greenhouses as sentinels against gas, since they are affected more quickly than humans. But Doctor Crocker finds that the tomato plant has an even more sensitive nose for gas than the canary bird. Even one part of gas to 100,000 parts of atmosphere is enough to make the leaves of the tomato plant turn downward.

Committee of Ten Discusses Possibilities of Silica Gel Air Conditioning with Coal

Important and valuable work was transacted by the Committee of Ten of the Coal and Heating Industries at its meeting, held in Cleveland, January 26, at the time of the meetings and exposition held in that city in connection with conventions of the heating and refrigerating engineers.

A report on the use of silica gel in connection with air conditioning was presented by Chairman J. H. Walker, of the research committee. He told of the activities of the gas industry in utilization of silica gel and pointed out that some research in this medium with solid fuels might bring developments of considerable importance. The subsequent discussion developed information that the heating and ventilating exposition at the Cleveland Auditorium contained the most recent equipment for air conditioning with appliances adapted to coal as well as other fuels.

A definite outline for a manual on solid-fuel heating was reported by Homer R. Linn, chairman of the manual committee. The program includes the publication in the language of the layman, but technically correct, of a series of bulletins on the various subjects involved in providing heating satisfaction, each bulletin to appear as soon as it can be prepared. It is proposed that when the series has been completed the bulletins be combined to form a complete manual on solid fuels and related equipment. The subjects now in the course of preparation include: chimneys, fireplaces, draft, combustion, coal, hand-firing, stokers, heat controls, warm-air heating, hot water heating, steam heating, air conditioning, comparative data on coal, oil and gas, certified heating, and correcting heating troubles. Bulletins on chimneys, fireplaces and certified heating are now practically completed, and several others are in preliminary stages of preparation.

The committee gave formal indorsement to the suggested ordinance for chimney construction, prepared by the National Board of Fire Underwriters, which already has been approved by the Heating & Piping Contractors National Association, the National Warm Air Heating Association, and the American Society of Heating and Ventilating Engineers, along with a number of other national organizations interested in proper building construction.

Demand for Single Family Houses

Less than 5% of single-family houses were vacant in two-thirds of the cities reporting in a recent survey of vacancies in residential buildings made by local organizations and reported to the Department of Commerce.

The survey, conducted in 37 cities, shows that single-family houses are more in demand than flats or apartments. In none of these cities did the vacancies exceed 9.5% of the single-family houses. Vacancies in apartments ranged from 6.8% to 27.2%; in flats from 4.4% to 20.2%, and in two-family houses from 3.4% to 20.1%. In no city were total vacancies in all types of residential property more than 11.2%, the survey shows.

The data were compiled by the Division of Building and Housing of the Department of Commerce from surveys made by local organizations in various cities.

Smoke Intensity Measured by Photoelectric Cell

A photoelectric recorder, so sensitive that it can record the intensity of smoke, was described by C. W. LaPierre, of the General Electric Company, at the opening session of

the annual winter convention of the American Institute of Electrical Engineers, held in New York, January 25-29.

Mr. LaPierre told how he had taken the record of smoke intensity in the stack of a steam generating furnace, using the photoelectric recorder in connection with a smoke detector. Other applications of the instrument include the securing of electro-analysis survey records, turbo-generator vibration recordings and small draft pressures.

"The photoelectric recorder possesses all the desirable characteristics of direct-acting recorders, and at the same time is capable of a sensitivity and rapidity of response equal to the highest sensitivity indicating instruments," he said. "In general, anything which can be indicated can now be continuously recorded by means of the photoelectric recorder."

N. Y. Academy of Medicine Holds Pure Air Show

The New York Academy of Medicine conducted an exhibit devoted to promotion of pure air and outdoor cleanliness, January 12-19, in New York. One section of the show dealt with air pollution and the other with outdoor cleanliness in relation to pure air. Exhibits in the air pollution section illustrated how air becomes polluted, the extent, the harmful substances in polluted air, methods of measuring impurities, and ways of preventing pollution. The outdoor section showed the extent of the general problem in various cities and methods of handling in this country and abroad.

Eicher Reports Change in School Code for Pennsylvania

Dr. Hubert C. Eicher, director of the school buildings division, Pennsylvania Department of Public Instruction, in an address before the Pittsburgh chapter of the A.S.H.V.E., February 15, stated that nine experts in different branches are revising the school code of Pennsylvania. It is presumed that heating and ventilating requirements will be among those changed.

Doctor Eicher said he considered misunderstanding to be chiefly responsible for the friction which exists among engineers, architects, school officials, and men in public office. Letters are too meager a medium of exchange of ideas to straighten out seeming difficulties, but when alleged enemies sit down together and talk things through, they usually come to satisfactory decisions. A school building code is adopted to insure the erection of economical and well-equipped buildings which will guard the health of boys and girls. Doctor Eicher traced the history of Pennsylvania school building codes, showing how they are applied and how they have been changed. In 1911, a law placed all plans and specifications of school buildings within the jurisdiction of the State Board of Public Education. There were 1400 *shall's* in this code, which was not perfect, but which served well until 1921, when it was revised. With a few slight changes in 1924, this revised code has served Pennsylvania admirably until now. The State Health Department reports that since 1921 the health of the boys and girls has been greatly improved.

Tapp Opposes Oil Embargo as Obstacle to Burner Industry

Harry F. Tapp, executive secretary of the American Oil Burner Association, who recently appeared before the House Ways and Means Committee, told the committee that the proposed embargo on petroleum and petroleum products would add thirty million dollars a year to the fuel bill of oil burner users and approximately half a billion dollars a year additional expense for all users of gasoline and petroleum products.

"The oil burner industry is not responsible for the condition existing in the petroleum industry to-

(Left) A nurse, employed in the construction of New York's $22,000,000 tubes under the East River, watching the steady increase in pressure in a compression chamber, in the offices of Mason & Hanger, contractors, where workmen are treated for "bends," air bubbles in the blood, caused by too sudden change from one pressure to another while at work. (Below) Two of the chambers, the one on left for passing material, and the one at right for workmen, in the construction of the new tunnel.

Photos by Ewing Galloway

day," Mr. Tapp told the committee. "On the contrary, were it not for the increased use of fuel oil during the past ten years, the petroleum industry would indeed be in a critical condition.

"Last year approximately 350,000,000 barrels of oil fuel were consumed in this country and more than 50,000,000 barrels were used for the heating of homes and commercial buildings. This oil is the by-product of refining operation. If it is not used as fuel, the problem of its disposal would be a serious one."

Mr. Tapp predicted that the proposed tariff or tax measure, if passed, would impose a real hardship on the oil burner industry. This would mean, in addition to the heavy financial losses, thousands of men added to the ranks of the unemployed and a slowing down in many allied lines of industry.

Mr. Tapp previously had forwarded to every member in both houses of Congress a resolution by the American Oil Burner Association going on record in opposition to the proposed tax.

A. S. H. V. E. Urges Members to Influence Ventilation Legislation

The headquaters of the American Society of Heating and Ventilating Engineers mailed a bulletin, dated January 22, to members of the society living in the state of New York, calling their attention to the fact that the committee on ventilation standards would report at the Cleveland meeting, and urging members to reflect to their representatives in the New York Senate their views of the matter.

The bill before the legislature of the state is Senate Bill 130, introduced by Senator Mastick as "an act to amend the education law in relation to plans and specifications for construction of school buildings." The bill is the same as that introduced in 1931, number 552, removing the present standard of ventilation, and leaving the question of specific requirements open by stating that plans "must insure proper ventilation."

The bulletin requests that members send a copy of any communication they have with their representative at Albany to the society headquarters.

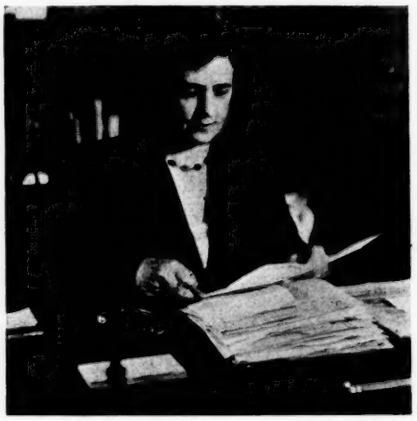
New York May Have New Building Code

On January 14, the Proposed Building Code for New York was presented to Mayor Walker by the Merchants Association. Both the mayor and the representatives of the association remarked on the importance of this accomplishment. The printed code which is now available for study is voluminous and exhaustive.

Prominent among the new features introduced is the one which recognizes a type of construction designated as "fire-protected." This type of construction takes note of new methods and materials and it is predicted that buildings erected in accordance with the provisions of the code will cost not less than 10% less than when construction follows the standards of the present code.

Other points of general interest are the provisions which permit lighter masonry, higher unit stresses in steel under certain conditions, increased elevator speeds, and the use of welding instead of riveting when the welding is done in accordance with prescribed standards.

The heating and ventilating provisions are contained in separate sections of the code. Both contain new features and differ sharply from those contained in the old code.



Katharine Fisher, director of Good Housekeeping Institute, who will address ninth annual oil burner convention.

The preparation of the code was accomplished through the work of sub-committees of the committee on building laws and regulations of the Merchants Association.

Clyde R. Place acted as the chairman of the sub-committee on mechanical and sanitary equipment, and also as chairman of the sub-committee on heating and ventilating. Other members of this sub-committee were: B. H. Belknap, W. H. Driscoll, Charles Hoffman, Arthur K. Ohmes and Frederic N. Whitley.

Detroit Group Condemns Free Engineering

A motion disapproving of the practice of manufacturers furnishing free engineering was passed by the Michigan chapter of the A.S.H.V.E. at its meeting held January 18. The Detroit Edison Company was host to the members.

The resolution read as follows:

"Resolved, that the Michigan chapter of the American Society of Heating and Ventilating Engineers place itself on record as disapproving the practice of manufacturers and sales agents of furnishing free engineering layouts and specifications to contractors and architects, and be it further resolved that those so doing be requested by the society to discontinue this practice, and that the secretary be instructed to send copies of this resolution to the national secretary of this society."

C. H. Hirshfeld, chief of the research department, Detroit Edison Company, addressed the meeting on What Next?, in which the speaker discussed the past and future developments of steam power plants.

Oil Burner Show to be Open to Entire Industry

A decision to enlarge the ninth annual oil burner show, to be held in Boston, April 11-16, was made at the quarterly meeting of the directors of the American Oil Burner Association, in Boston, recently. Heretofore, the show has been open only to members of the A.O.B.A. Under the new plan the membership

requirement for accessory or associate exhibition is waived. The privilege of exhibiting oil burners will be confined to members as heretofore.

H. & P. C. N. A. Appoints Convention Committee

Preparations for the forty-third annual convention of the Heating and Piping Contractors National Association, to be held in Detroit, May 16-18, have begun with the naming of a convention committee by the members of the Detroit local association. The committee will have charge of all arrangements and plans for the convention.

George Freyn, Detroit heating contractor, has been named chairman of the convention committee. He will have as members of his committee the following assistants, each of whom will head the sub-committee indicated:

Emmet Byrne, reception committee; Charles Gallarno, publicity; Clare Davis, finance committee; Emery Harris, entertainment committee; Mrs. Ray Spitzley, ladies' committee; Edward Glanz, golf tournament committee; and Lawrence McConachie, hotel and transportation committee. George S. Clarke, executive secretary of the Heating and Piping Contractors Detroit Association, will work with the general committee as an ex-officio member.

Propeller Fans Save Crops from Frost

Growers of citrus fruits in Texas are using giant propeller fans to carry air currents across the fields to prevent frost damage to crops, according to Associated Press dispatches from that state. It is claimed that this method is much cheaper than the use of smudge pots.

Grand Rapids Engineers Discuss Reversed Refrigeration Cycle

A review of A. R. Stevenson's A.S.R.E. paper *Application of Refrigeration to Heating and Cooling of Homes,* read at the Cleveland convention, was presented by Prof. H. B. Dirks, Dean of Engineering, Michigan State College, at a business meeting of the Western Michigan chapter, held February 8, at the Rowe Hotel, Grand Rapids. *Changes in Ionic Content of Air in Occupied Rooms Ventilated by Natural and Mechanical Means,* a paper presented by C. P. Yaglou at the Cleveland convention, was summarized by Prof. L. G. Miller, also of Michigan State College.

Preceding the discussions of papers, President O. D. Marshall, who officiated, gave a brief resume of his visit to the Cleveland convention.

Describe Banks for Discount of Loans on Homes

The bill providing for the establishment of 12 Federal home loan banks to discount residential home mortgages, which was introduced in the Senate by Senator Watson, and in the House of Representatives by Representative Luce, of Massachusetts, in December, does not differ from the plan outlined by President Hoover on November 13, when he first proposed the new system, according to the National Association of Real Estate Boards.

The discount banks would not deal with individuals.

The discount banks would have no relation whatever with the Federal Reserve. Thus, no broadening of the Federal Reserve base is contemplated in this bill.

The 12 banks would be located in the districts already defined for the Federal Land Bank system.

As provided in the bill, the new banks would operate as follows:

Amortized and unamortized mortgages would be discounted by the banks. All mortgages handled must be on dwellings designed for not more than two-family occupancy.

Discounting of unamortized loans will be restricted to 50% of the unpaid principal amount of the loan, also restricted to loans having less than five years to run. For example, if the unpaid balance of the principal is $4,000, the maximum loan would be $2,000, providing for possible depreciation.

Amortized mortgages will be discounted up to 60% of the unpaid amount, when the original term is not more than ten years.

Only mortgages having an unpaid balance of less than $15,000 will be eligible for discount.

The discount banks will deal with their members who are defined in the bill as "banks and banking associations, which would include savings banks, trust companies, insurance companies, building and loan associations, and other institutions subject to the inspection and control of banking laws."

The discount banks will keep themselves liquid and in cash through the sale to the public of tax exempt notes, debentures or bonds, which will be secured by the mortgages they have purchased. A margin between the interest rate charged member borrowers for their loans and the rate paid by the discount banks on securities they issue is to provide operation expenses and reserves.

The reserve accounts will be built up from net earnings and when they have reached a certain sum provisions are made for the payment of dividends.

Each of the banks may resell advances on mortgages to any of the other discount banks. This, comments the association, will provide the flow of capital for home buying and building from state to state that is greatly needed.

Air Conditioning Subject of Chicago Meeting

Edwin A. Jones, president of the Milwaukee chapter, A.S.H.V.E., addressed the Illinois chapter at its meeting, held February 8, at the Hotel Sherman, Chicago, on the subject of heating and air conditioning of homes. Among the topics receiving consideration were the need of forced circulation, and the advisability of recirculation, the speaker expressing a preference for from five to six changes of air per hr.

Mr. Jones told how conditions of comfort and health may be attained in homes, stressing the methods of insuring optimum conditions as to

air purity, temperature, humidity, and circulation.

Regarding duct work, the introduction of air 8 ft. above floors was advocated, with a system of return ducts. Accessible dampers were recommended for all ducts. Quiet operation was considered a primary requirement for residential work, this being largely dependent upon the type of fan installed. High duct velocities were held not to be troublesome as noise producers.

Humidification in winter and dehumidification and cooling in summer were discussed. In the opinion of the speaker, air washers are considered good humidifiers but are not entirely satisfactory for air cleaning. However, the convenience of air washers was commended. With some of the newer humidifying apparatus, double glazing and house insulation are necessities in preventing trouble from condensation. Commenting on thermostats, Mr. Jones stated that they should be located at the maximum distance from the furnace.

Other topics which received attention were designing for adequate circulation, bringing outside air into the house for summer cooling, use of ice as a cooling medium, and use of cooling water from an artesian well. The speaker urged competent professional advice for the homeowner in order to forestall acceptance of cheap bids on suitable appliances. Mr. Jones recommended the use of the following velocities: ducts, 1000 ft.; risers, 600 ft. to 700 ft.; registers, 600 ft. per min.

President J. J. Aeberly officiated at the business meeting, and John Howatt rendered a report on his visit to the Cleveland convention.

London Concerned with Smoke Problem

In his address to the annual conference of the National Smoke Abatement Society, England, the president, Dr. H. A. des Voeux, said that excessive smoke was not a necessity, but a luxury of laziness, which should not be allowed by authorities charged with the protection of the health of the people, according to the London correspondent of *The Journal of the American Medical Association.*

"The local authorities were working in conjunction with the Department of Scientific and Industrial Research, which was investigating the impurities of the atmosphere," he says. "In the square mile of the city of London, 150 tons of dirt fell in 1911, while in 1930 the figure was 380 tons. Similar figures were obtainable from other large centers. The destructive element in the combustion of coal was now known to be sulphur dioxide, which was turned into sulphuric acid in the air. The report on the disastrous fogs in the Meuse Valley last winter stated definitely that the deaths were due to suffocation, which might easily be reproduced in closely inhabited neighborhoods had not the government been forewarned in time to insist that measures must be taken to prevent the escape of this gas into the air, or to neutralize it before it escaped.

"At the new electric station being erected at Battersea, in London, the enormous amount of 2,000 tons of coal would be burned daily, emitting at the smallest calculation 30 tons of sulphur, which is equivalent to 90 tons of sulphuric acid. Imagine the effect of this acid accumulating as it well might, for four or five days on the unfortunate population of closely packed Battersea.

"A report was presented by the Greater London Joint Smoke Abatement Committee, regretting that no satisfactory method of measuring the density of the smoke had yet been evolved. The subject was still receiving the attention of the government fuel research station. Great progress had been made in London in connection with the control of smoke emission. Beneficial effects were being secured under the law which prohibited the emission of black smoke for more than three minutes in thirty, but the committee was still not satisfied and regarded a two-minute period as necessary, which it hoped would be in force in 1936."

Noise Standards to be Formulated by A. S. A.

The organization of a sectional committee on noise measurement under the procedure of the American Standards Association was recommended by a general conference of 32 representatives of 18 national bodies, held in New York, January 29. It was also recommended that the committee function under the technical leadership of the Acoustical Society of America.

The scope of the project as recommended by a committee consisting of Prof. Vern O. Knudsen, Acoustical Society of America; E. E. Free, American Society of Mechanical Engineers; P. L. Alger, American Institute of Electrical Engineers; H. R. Summerhayes, National Electrical Manufacturers Association; R. G. McCurdy, ASA Telephone

Courtesy The New York Sun

New York and New Jersey Smoke Abatement Board
(Left to right) Dr. H. H. Sheldon, Laura A. Cauble, Jacob H. Haffner, and Health Commissioner Shirley W. Wynne.

Group; and Dr. Harvey Fletcher, Acoustical Society of America, is as follows:

> Preparation of general standards of nomenclature, units, scales, and measurement in the field of acoustics, with special reference to noise measurements.

The conference followed a request made by the Acoustical Society of America to the American Standards Association to set up a national committee to correlate the activities of various technical and trade organizations which have been attempting individually to set up standards for noise measurement. Eight such organizations now have committees on this subject. It was pointed out that the diverse nomenclature and methods of measurement growing out of these uncorrelated activities were interfering with the progress of scientific work on the subject, and that without real national standardization this condition would become worse as different individuals became accustomed to different concepts and terms.

For the present, the committee's work will be concentrated chiefly on the standardization of nomenclature, units, and scales. It is believed that further progress in the science of noise measurement is necessary before effective standardization of this phase of the subject can be completed. There is some confusion in the measurement of noise at the present time because of the complex nature of noises and their effect upon the ear, and the fact that none of the noise meters yet designed can translate the approximate physiological and psychological effects of noise into definite units of measurement. Dr. Harvey Fletcher, of the Bell Telephone Laboratories, pointed out at the conference that to the average individual a noise composed of widely separated frequencies is less disturbing than a noise of equal intensity but having components close together in the frequency range.

Mild Weather Causes 25%–30% Decrease in Fuel Consumption

Mean monthly temperatures for the heating season from September 1 to January 31, have averaged from 1.8° to 11.9° above normal in five leading cities in the country, resulting in an estimated drop in fuel consumption for the heating of buildings of 25% to 30%.

Table 1, prepared by G. J. O'Connor of the New York Weather Bureau, shows the mean monthly temperatures for the five cities and the amount this temperature is above normal. Table 2 shows the number of degree days for the season September 1, 1931 to January 31, 1932, and also shows that the number of degree days this heating season ranges from 25.0% to 31.9% below normal. Boston has the smallest departure from normal, so far as degree days are concerned, while Pittsburgh leads.

Inasmuch as the degree day is a fairly accurate method of measuring fuel consumption, the loss in volume for fuel companies doing business in heating is in direct proportion to the percentage shown in column 3 of Table 2.

Table 1
MONTHLY MEAN TEMPERATURES AND THEIR DEPARTURE FROM NORMAL
FIVE CITIES

	New York		Boston		Pittsburgh		Chicago		St. Louis	
	Monthly Mean Temperature	Degrees Above Normal	Monthly Mean Temperature	Degrees Above Normal	Monthly Mean Temperature	Degrees Above Normal	Monthly Mean Temperature	Degrees Above Normal	Monthly Mean Temperature	Degrees Above Normal
September	71.2	4.4	66.9	3.6	70.3	3.9	71.2	6.4	76.9	6.4
October	60.4	4.1	58.5	4.9	57.4	1.8	58.8	4.8	62.9	4.1
November	51.5	7.3	49.4	7.4	51.6	8.4	50.0	9.9	54.8	9.4
December	40.6	5.6	36.6	4.1	41.5	7.3	38.4	9.6	45.2	10.3
January	42.8	11.9	38.8	10.9	41.8	11.3	33.6	9.9	39.8	8.7

Table 2
DEGREE DAYS — SEPTEMBER TO JANUARY — FIVE CITIES

	Degree Days Sept. 1 to Jan. 31 This Season	Degree Days Sept. 1 to Jan. 31 Normal Year	Per Cent This Year Below Normal
New York	2049	2910	29.6
Boston	2460	3280	25.0
Pittsburgh	2179	3003	27.4
Chicago	2515	3514	28.4
St. Louis	1904	2798	31.9

Better Heating Association Formed in Pittsburgh

Local support of the activities of the Committee of Ten of the Coal and Heating Industries was formally inaugurated in Pittsburgh through organization of the Better Heating Association of Western Pennsylvania, at a meeting of representatives of solid fuel and related equipment industries at the Keystone Athletic Club, Pittsburgh, January 28.

A board of directors, consisting of one member representing each basic class, was elected for every class with the exception of stokers. The stoker member of the board will be elected at a later date.

The classifications tentatively approved and the member of the board elected to represent it are: coal producers, John C. Hoffstot, Lincoln Gas Coal Company; heating and piping contractors, Peter O'Neal, Bartley O'Neal Company; warm-air heating, W. J. Keist, W. J. Keist and Son; railroads, J. B. McCorkle, Pennsylvania Railroad; heating accessories, John H. Wagner, Wagner-Proie Furnace Company; coal and coke wholesalers, C. S. B. Ward, Weiman & Ward, and president of the American Wholesale Coal Association; boiler and radiator manufacturers, W. A. Bowman, American Radiator Company; sheet metal contractors, E. W. Riesmeyer, E. H. Riesmeyer Company; coke producer, A. J. Miller, Davis Coal & Coke Company; coal and coke retailers, H. W. Kiser, Alex Black Coal Company, and

president of the Pittsburgh Retail Coal Merchants Association; heating and ventilating engineers, F. C. Houghten, research director of the A.S.H.V.E.; automatic heat controls, E. G. Smyers, T. F. Campbell Company.

John C. Hoffstot was elected president of the organization; W. J. Keist, vice-president, and H. Kennedy Hanse, secretary-treasurer.

Use of Oil Doubles During Past Decade

The use of fuel for power by American manufacturing industries practically doubled in the decade from 1919 to 1929, according to a recent report of the Bureau of Census. Compared with 1919, the government figures show increases in 1929 in the use of all kinds of fuel except anthracite coal.

Approximately $1,500,000,000 was spent by manufacturing interests for fuel in 1929, of which $220,195,713 went for petroleum products, including gasoline and kerosene. Fuel oil used in oil burners was responsible for $212,639,372, representing 6,583,198,995 gal. In the same year approximately 972,174,000 gal. of fuel oil were used in 535,000 domestic oil burners in American homes, the Oil Heating Institute announced.

Winnipeg District Heating Plant Described at Toronto Meeting

The district heating system of the city of Winnipeg, regarded as one of the most up-to-date plants on the continent, was described by G. W. Oliver, of the Winnipeg Hydro-Electric System, at the forty-sixth annual meeting of the Engineering Institue of Canada.

The principal reason for the establishment of the plant, the speaker said, was the desire to utilize the existing boilers of the steam-electric stand-by plant during the winter months, in this way obtaining additional revenue. The equipment of the station includes four water-tube boilers fired with pulverized coal and having a combined capacity of 440,000 lb. of steam per hr. at 250 lb. pressure. There are also three electric boilers, each taking 7500 kw., with a total capacity of 62,000 lb. of steam per hr.

These boilers serve to flatten out the load curve of the hydro-electric system by using off-peak energy, in this way substantially increasing the load factor of the hydro-electric system. The three turbine-generators, having a total rating of 11,000 kw. exhaust to condensers and the steam they use is not employed in the heating system, which is supplied by live steam both from the electric and coal-fired boilers. The city steam distribution system consists of a loop main and looped branches, laid out in this way with a view to eliminating dead ends. The mains range in size from 4 to 14 in. in diameter. They are laid in concrete conduit and expansion is provided for by use of slip joints.

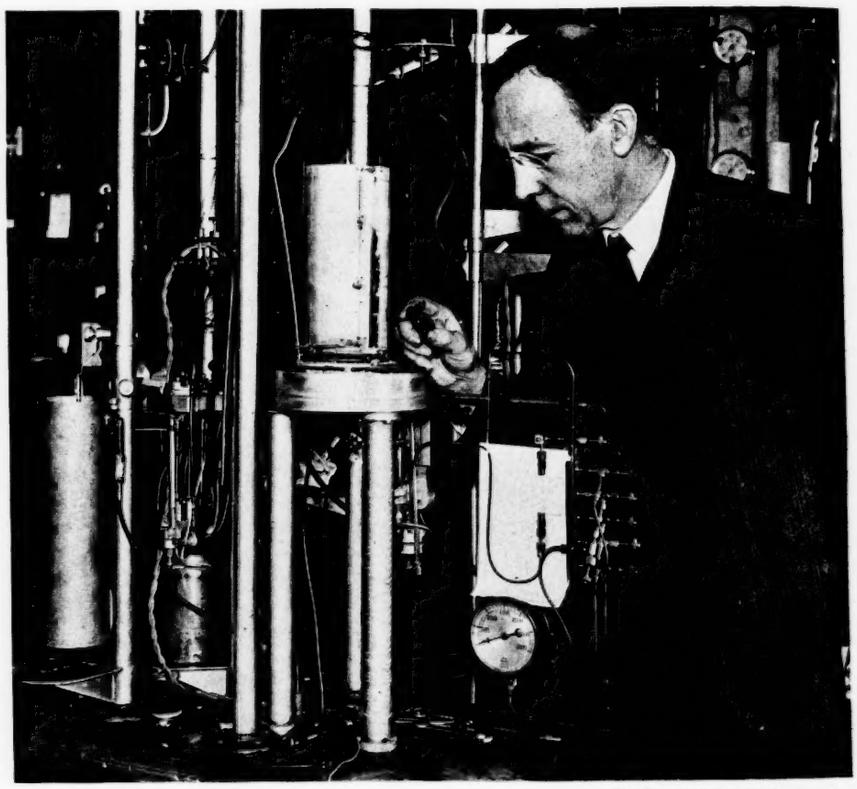
The heating system supplies the business area of Winnipeg, and 246 buildings are at present connected. The steam consumption ranges from 250 lb. to 650 lb. per sq. ft. of radiation per season. Steam reaches the various buildings at a pressure of approximately 40 lb., which is reduced on the customer's premises to suit conditions.

1931 Was Warmest of 63 Years in New York

The year 1931 was the warmest on record in New York City in 63 years, was very clean atmospherically and was unusually dry, according to the sixty-third annual report of the New York Meteorological Observatory in Central Park, recently made public by David R. Morris, meteorologist in charge.

The wind blew 77,455 miles in 1931, at an altitude of 62 ft. above the ground, which is an average hourly velocity of 8.8 miles, 0.8 less than normal. March was the windiest month, with an average velocity of 11.9 miles, and July was the least windy, with 6.7 miles average hourly velocity.

March 17, St. Patrick's Day, shared with April 1, All Fools' Day, the distinction of having been the windiest days, with an average hourly velocity of 19.4 miles. The highest velocity for a five-minute period was 44 miles an hour from



Underwood & Underwood

Dr. N. S. Osborne of the Bureau of Standards at Washington investigating the properties of superheated steam at very high pressures, using a newly-developed steam calorimeter which can be used under extraordinarily high pressures and temperatures.

the east-northwest on April 1, and a wind gust of 64 miles occurred on that date.

The average daily temperature for 1931 was 55.4°, or a daily excess of 2.9° over the normal. The unusual warmth of the year was described as a "steady month-to-month warmth" rather than one characterized by spurts of high temperature. Every month of the average daily temperature was above normal, and from September on the abnormal warmth was accentuated.

July, with an unofficial average of 99° in the sun, was the warmest, and January, with 52°, was the coolest month. The highest temperature in the sun was 137 in June, July and August.

Builders Cut Wages in Large Centers

Wage cuts in the building trades in the whole country are foreseen as a result of action taken by builders in New York and Chicago in establishing wage rates from 20% to 30% below existing scales. The unions in New York have not formally accepted the cuts, but the Building Trades Council, Chicago union organization, has accepted a 20% decrease. New York builders have asked for a 25% drop in union scales, and it is expected that a 15% or 20% drop will be accepted by the workers.

Representatives of thirty trade groups in the Building Trades Employers' Association of New York City formally ratified a new wage schedule effecting cuts of 25% to 30% to go into force May 1, when existing agreements expire.

The revised schedule, which will apply to all work for the 20-month period ending on December 31, 1933, will affect about 115,000 organized workers in the industry. It was approved unanimously at a lengthy meeting of eighty executives and committee members of the association, held at their headquarters.

The approval constituted final action by the employers on their own initiative, and members will be notified to use the new standards in their future contracts, as they do not expect to receive or consider any further counter-proposals from labor officials, Christian G. Norman, chairman of the board of governors of the employers' association, announced after the meeting.

Mr. Norman said that no last-minute proposal had been offered by the labor men and that no representatives of labor had attended the meeting. The association acted independently after negotiations with the Building Trades Council had resulted in a deadlock, the labor men having refused to sign an agreement for reductions amounting to as much as 25%, and the employers having rejected several weeks ago the workers' suggestion for a cut of slightly less than 10%.

Mr. Norman explained that the length of the session, almost three hours, had been caused not by any lack of unanimity among the association members but by a discussion of other questions, including the proposed State law to license contractors, and other legislative matters. Each trade made a report supporting the plan of the board of governors for the wage cut, he said.

"It is our belief that the scale unanimously adopted by our members as a fair wage for the building trades will do much to stabilize conditions in the industry and to eliminate the bootlegging which has been going on for the last year under wages averaging much lower than the scale we have approved," Mr. Norman said, explaining that for some time outside interests had disrupted the trade by underbidding those who adhere to the existing schedules.

"Since about 50% of the cost of constructing the average building goes for labor, it is our hope that the lower rates will serve to foster new building work, particularly in public works, utilities and institutional classifications, where expansion programs have been held up for some time."

The following table shows the new rates and those now in effect in some classifications:

	May 1 Rate.	Pres. Scale.
Asbestos workers and insulators	$10.00	$13.20
Asbestos workers' and insulators' helpers	7.00	9.90
Plumbers	10.00	13.20
Sheet metal workers	10.00	13.20
Steam and hot water fitters	10.00	13.20
Steam and hot water fitters' helpers	7.00	9.90

The Construction Employers' Association of Chicago, which had asked for a 25% cut in the union scale, and the Building Trades Council, representing 150,000 workmen, agreed to a cut of 20%, the workers submitting to the wage slash in the hope that it would stimulate building. This was done in spite of the fact that most of the workmen have agreements which do not expire until 1934.

In Portland, Ore., a reduction of

COMING EVENTS

MARCH 9-10, 1932. Spring Meeting of the Eastern Supply Association, to be held at the Hotel Astor, New York.

APRIL 11-16, 1932. Ninth Annual Convention and Oil Burner Show of the American Oil Burner Association, Mechanics Building, Boston, Mass.

APRIL 14-15, 1932. Fifth Midwest Bituminous Coal Conference, Purdue University, Lafayette, Ind.

APRIL 20-30, 1932. National Home Building and Home Ownership Exposition, to be held at the Grand Central Palace, New York.

APRIL 27-29, 1932. Annual Meeting of the American Welding Society, Engineering Societies Building, New York.

MAY 9-11, 1932. Annual Meeting of the Natural Gas Department, American Gas Association, to be held in Tulsa, Okla. Headquarters at the Mayo Hotel.

MAY 16-18, 1932. Forty-Third Annual Convention of the Heating and Piping Contractors' National Association, to be held in Detroit, Mich., Headquarters at the Book-Cadillac Hotel.

MAY 23-25, 1932. Annual Convention of the American Boiler Manufacturers Association, to be held in White Sulphur Springs, W. Va. Headquarters at the Greenbrier Hotel.

JUNE 6-10, 1932. Twenty-Fifth Annual Convention and Exposition of the National Association of Building Owners and Managers, to be held in Chicago. Headquarters at the Drake Hotel.

JUNE 14-17, 1932. Annual Convention of the National District Heating Association, to be held in Pittsburgh, Pa. Headquarters at the William Penn Hotel.

JUNE 20-23, 1932. Fiftieth Annual Convention and Exhibit of the National Association of Master Plumbers, at Madison Square Garden, New York.

JUNE 26-30, 1932. Semi-annual Meeting of the American Society of Heating and Ventilating Engineers, to be held in Milwaukee, Wis.

JUNE 27-JULY 1, 1932. Semi-annual Meeting of the American Society of Mechanical Engineers, to be held in Bigwin, Canada.

News of the Month

20% was voted by the arbitration board of nine on February 4. By the decision the reduction is made retroactive to January 2.

The Associated Building Interests, St. Louis contractors' organization, has asked the Building Trades Council, union organization with 18,000 members, to accept a 33 1/3% reduction.

Eames Opens Office

D. D. Eames, consulting engineer, has opened an office at 410 Commonwealth Ave., Boston, Mass. For more than 21 years, Mr: Eames has been connected with Lockwood Greene Engineers, Inc., in responsible charge of all power plant design, heating, ventilating, fire protection, special reports and problems of many kinds. He will continue to specialize in this field under his own name.

W. G. Hollinworth

W. G. Hollinworth, formerly secretary of the Institution of Heating and Ventilating Engineers, England, died December 12 at the age of 75.

Mr. Hollinworth became secretary of the institution in 1909, a position he held until 1924, when he had a stroke which prevented him from carrying on active business. His daughter carried on until he retired in 1926.

In 1920 he also became secretary of the Institute of British Foundrymen and the British Acetylene Welding Association.

Mr. Hollinworth was European representative of HEATING AND VENTILATING since the magazine was founded in 1904.

Raymond Newcomb

Raymond Newcomb, vice-president, and E. N. Cunningham, Birmingham representative, of the Fitzgibbons Boiler Company, Inc., and the Kewanee Boiler Company, Inc., died as a result of injuries received in an automobile accident on Feb. 15 near Birmingham.

Mr. Newcomb was 34 years old. He was a graduate of Massachusetts Institute of Technology and did post-graduate work at Stevens Institute, specializing in boiler design and operation. During the war he held an ensign's commission and had charge of boiler operation aboard the large transports. At the close of the war he became associated with the two boiler companies and was rounding out thirteen years of service, successively, in the shop, as a salesman, as manager of the Boston office and finally as vice-president in charge of branch sales.

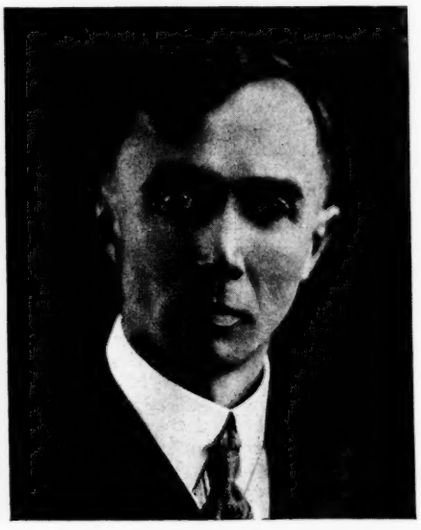
Mr. Newcomb was deeply interested in boiler development and his service to the industry in this direction is exemplified by his active participation in the work of the following societies and committees of which he was a member: A.S.M.E. boiler code committee, heating boiler division; Steel Heating Boiler Institute, boiler code committee; A. S. H. V. E. committee on rating and testing of low pressure boilers, and member of the board of governors of the New York chapter of the A.S.H.V.E.

Through his business relations and society activities he was widely known throughout the industry and the trade, and was liked and respected for his earnestness, sincerity, straight dealing and thorough technical and practical knowledge. His passing leaves a gap in the active ranks of the boiler industry that will not be easily filled.

Cleveland Meetings and Exposition Focus Spotlight on Air Conditioning

Air cooling, cleaning, humidifying and dehumidifying, and air conditioning in general, held the center of the stage at the Second International Heating and Ventilating Exposition and the annual meetings of the American Society of Heating and Ventilating Engineers and the American Society of Refrigerating Engineers, held in Cleveland, January 25-29. Well over 700 members registered at the A.S.H. V.E. meetings alone.

Notable features of the exposition were the domestic air conditioning units, dehumidifying units for residential uses, a complete line of room coolers, and several new instruments for measuring humidity. Several new unit heaters were displayed, improved heating specialties were on exhibition, and the control group showed several new lines. One company displayed a complete new line of boilers and radiators, and others some unusual pipe insulation materials, while still others showed new types of filters and grilles.*



F. B. Rowley, new president of A. S. H. V. E.

A.S.H.V.E. Meetings

The thirty-eighth annual meeting of the American Society of Heating and Ventilating Engineers officially opened January 25. The technical sessions, however, did not begin until January 26.

In the report of the committee on research, Chairman C. V. Haynes said that cooperative research had been arranged for by contract with the University of Minnesota, University of Illinois, Harvard School of Public Health, University of Wisconsin, Yale University, Carnegie Institute of Technology, Agricultural and Mechanical College of Texas, and the Armour Institute of Technology. Among the general projects, either under investigation or consideration, are those on heat transmission, building materials, pipe

*Many of the new devices displayed are shown on pages 81-88.

sizes, air cleaning, air flow through grilles, radiator testing, garage ventilation, air conditioning and its relation to comfort, oil burning, sound and noise prevention, infiltration in buildings, and pumps and traps.

The committee reported that the committee on research has taken over the card files and physical properties of the Association for Correlating Thermal Research.

F. B. Rowley and W. A. Eckley presented a paper at the Tuesday morning session on *Surface Coefficients as Affected by Direction of Wind*[1].

F. C. Houghten and Carl Gutberlet described the research at the society's laboratory on *Conductivity of Concrete*[2].

R. A. Miller and L. V. Black presented a paper on *Transmission of Radiant Energy Through Glass*[3].

At the afternoon session, January 26, E. R. Queer stressed the *Importance of Radiation in Heat Transfer Through Air Spaces*[4].

F. C. Houghten and Carl Gutberlet described the *Heat Emission from Iron and Copper Pipe*[5].

The paper, *Supplementary Friction Heads in One-Inch Cast-Iron Tees*[6], was presented by title only, as F. E. Giesecke and W. H. Badgett, the authors, were not present.

A. R. Mumford presented the paper prepared by him and Dr. R. E. Hall, *Some Fundamental Considerations of Corrosion in Steam and Condensate Lines*[7].

Changes in Ionic Content of Air in Occupied Rooms Ventilated by Natural and by Mechanical Methods[8] was presented by C. P. Yaglou, L. Claribel Benjamin and Sarah P. Choate.

One of the most popular papers was that by Prof. Vern O. Knudsen, of the University of California, Los Angeles branch, on *Acoustical Problems in the Heating and Ventilating of Buildings*[9].

Heat Transmission as Influenced by Heat Capacity and Solar Radiation[10], was presented by F. C. Houghten, J. L. Blackshaw, E. M. Pugh, and Paul McDermott, all of the research laboratory.

J. H. Walker, S. S. Sanford, and E. P. Wells, described their *Field Studies of Office Building Cooling*[11].

Thursday afternoon the report of the committee on testing and rating unit ventilators was presented. Considerable discussion resulted, especially regarding the use of the orifice to be used in measuring the output. The motion was made and passed that the code be adopted, effective January 1, 1934.

The committee on ventilation standards presented the report which was the result of the work the committee has done during the past year. The report was divided into a preamble and five sections, the sections relating to temperature and humidity, air quality, air motion, air distribution, and air quantity. The requirements called for were more or less general in wording, with the exception of the requirements that the air velocity in a room should be not more than 50 ft. per min., and that not less than 10 c.f.m. per occupant of a total air circulation should be taken from an outdoor source.

After considerable discussion, it was moved and passed that the report should be received but not accepted, and that further action should be taken by the society at its semi-annual meeting next summer.

Prof. L. E. Seeley, of Yale University, presented *A Study of Intermittent Operation of Oil Burners*[12], the result of cooperative research being carried on by the university between the A.S.H.V.E. and the A.O.B.A.

A. H. Barker received prolonged applause at the close of the presentation of his paper *Room Warming by Radiation*[13], which Mr. Barker came from England to present.

A Study of the Combustible Nature of Solid Fuels[14] was the title of a paper presented by R. V. Frost.

At the annual banquet, January 28, Dr. W. E. Wickenden, president of the Case School of Applied Science, was the speaker of the evening. The banquet was also the occasion of the first presentation of the F. Paul Anderson medal, awarded to Willis H. Carrier for distinguished scientific achievements in the field of heating, ventilating and air conditioning. The presentation was made by past president L. A. Harding.

Officers elected by the A.S.H.V.E. for the coming year are as follows: President, F. B. Rowley; first vice-president, W. T. Jones; second vice-president, C. V. Haynes; treasurer, F. D. Mensing; newly-elected members of the council are F. E. Giesecke, G. L. Larson, J. F. McIntire, and W. E. Stark; newly-elected members of the research committee are D. E. French, F. E. Giesecke, L. A. Harding, and A. P. Kratz.

A.S.R.E. Meetings

Several of the sessions of the twenty-seventh annual meeting of the American Society of Refrigerating Engineers were not of primary interest to heating and ventilating engineers. On the other hand, the joint session Thursday morning included *Application of Electric Refrigerators to the Heating and Cooling of Houses*[15], by A. R. Stevenson, Jr., F. H. Faust, and E. W. Roessler.

Claude A. Bulkley presented a new psychrometric chart and a paper on *Air Conditioning as Applied to Cold Storage*[16]. The author's new chart covers a range from —40° dry-bulb temperature to 80° dry bulb.

Other papers of interest to heating and ventilating engineers were *Properties of Metal Foil as an Insulating Material,* by J. L. Gregg; *Summer Air Conditioning of Residences,* by Eugene D. Milener, and *Survey of Ice for Air Cooling Applications,* by R. T. Brizzolara.

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[1] Abstracted on page 71, HEATING AND VENTILATING, December, 1931.

[2] Abstracted on page 76.

[3] Abstracted on page 79.

[4] Abstracted on page 71, HEATING AND VENTILATING, December, 1931.

[5] Abstracted on page 63, HEATING AND VENTILATING, February, 1932.

[6] Abstracted on page 63, HEATING AND VENTILATING, February, 1932.

[7] Abstracted on page 61, HEATING AND VENTILATING, January, 1932.

[8] Abstracted on page 73, HEATING AND VENTILATING, November, 1931.

[9] Abstracted on page 75.

[10] To be abstracted next month.

[11] Abstracted on page 76.

[12] Abstracted on page 80.

[13] For complete paper see HEATING AND VENTILATING, February, 1932, page 48, and page 46, this issue.

[14] Abstracted on page 80.

[15] For complete paper see HEATING AND VENTILATING, February, 1932, page 31.

[16] Abstracted on page 74.

Abstracts . . .

Reviews and Summaries of . . . .
Current Papers, Books and Pamphlets

Air Conditioning Applied to Cold Storage, and a New Psychrometric Chart

By Claude A. Bulkley

A.S.R.E. paper, annual meeting, Cleveland, January, 1932. Approximately 4000 words, a graph with wet bulb temperature lines plotted on coordinates of dry-bulb temperature and specific heat, and a large working size reproduction of the Bulkley chart.

For use with cold storage applications, the author has developed a psychrometric chart for dry bulb temperatures between —41° and 80° F. The chart is so plotted that the reading interval at the low temperature is large enough to permit accurate readings of the properties. The chart follows closely the earlier one prepared by the author, but refinements and revisions have been made, simplified formulas are used in its calculation, and it is extended to lower temperatures. Two features of the chart are that the saturation line is plotted as a straight line and that the relative humidity lines are also straight.

The author feels that the application of the general ideas of air conditioning at present leaves much to be desired. Temperature has been too greatly stressed in cold storage practice to the exclusion of other factors of which humidity is an example. The advantage of low temperature as a means of retarding bacterial growth in stored food products, for instance, has been dominant. The fact that drying of such products usually results in the loss of some valued characteristic is well known.

As the loss of moisture of a substance increases in relatively dry air and decreases in moist air, the author calls attention to the desirability of surrounding many stored food products with very moist air at low temperatures. This will prevent loss of weight during storage which often not only results in deterioration but in loss of profit as well.

To maintain a high relative humidity in the space, the proper procedure is not to use methods of maintaining a low temperature which also extract moisture. Instead, the high moisture content should be automatically maintained using a cooling procedure which permits so doing.

The author feels that bunker coils are undesirable as a means of cooling cold storage rooms where high humidities are to be maintained since they remove moisture, as evidenced by the formation of frost on the coils, and that the objections urged against them also apply to dry coils located in fan coolers. The practical way to carry the desired high humidity and keep it under control is to pass the air through a spray chamber and to eliminate excess moisture. He proposes the use of brine as the spray liquid. Either the central or unit method may be used.

Regarding the use of a spray unit method of air conditioning for cold storage rooms, the author gives the following list of twelve advantages in favor of such a method:

1. Definite control of the relative humidity.
2. By carrying high relative humidity, shrinkage of food products is greatly reduced, with resultant improved quality of produce, particularly as applied to all kinds of meats and fruits.
3. On account of the very high heat transfer rate ($K = 140$ B.t.u. per deg. F. per hr. per sq. ft.) between the refrigerant and the brine spray, a relatively small amount of coil is required.
4. On account of this high transfer rate between the refrigerant and brine spray and the immediate absorption of heat from the air, drawn through the sprays, the refrigerating capacity of the brine spray type of unit is double that of the dry coil unit of the same size, and requires only 25% as much surface in the coil with the same difference between room and refrigerant temperatures.
5. Conversely, the same capacity can be obtained with half the temperature difference between room and refrigerant temperatures, thus materially raising the back pressure and reducing the size of compressor required.
6. The relative narrow range in temperature difference between room and refrigerant is shown by the following table for maintaining different relative humidities in the storage room.

Relative Humidity %	Temperature Difference Room and Refrigerant °F.
80	9.6
75	11.5
70	13.6
65	15.9
60	18.4

7. Much more rapid cooling of warm stock occurs when it is first brought into the room. Since this live load, during the cooling period, may be several times the dead load required to hold the stock for the longer storage period, the following table shows how the capacity of the unit goes up with rise in room temperature due to this live load. The table assumes that 80% relative humidity is to be maintained after cooling is accomplished and during the storage period.

Refrigerant Temp. °F.	Room Temp. °F.	Relative Capacity
25.4	35	1.00
25.4	40	1.78
25.4	45	2.56
25.4	50	3.34

If the compressor is of sufficient capacity to carry this increased load the relationship between refrigerant,

room and stock temperatures, will be automatically adjusted by a gradual rising of the back pressure on the compressor. Thus, the greatest possible extra capacity of which the equipment is capable will be brought into use for the peak load of cooling the warm stock.

8. Brine spray units give very uniform temperature throughout the room, due to rapid circulation of the air and low temperature difference between the room and air discharged from the outlets.

9. No frosting troubles can occur, the units operating at full capacity at all times.

10. Since, on account of the high relative humidity, less moisture will be removed from the stock, less will be condensed out of the air by the units, and since it never freezes in the brine spray, the latent heat of freezing is removed entirely as a part of the refrigeration load.

11. The units are portable, that is, can be easily disconnected and placed in new location.

12. As compared with bunker coils, less head room is required.

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Acoustical Problems in the Heating and Ventilating of Buildings

By Vern O. Knudsen

A.S.H.V.E. paper, annual meeting, Cleveland, January, 1932. Approximately 4000 words, six diagrams and graphs.

Lists the requirements for good hearing in a room, discusses the ideas of loudness of sound and its measurement, mentions briefly the effect of room form on the acoustics, the effects of reverberations and noise on the hearing of speech, gives tolerable limits of the amount of noise in various kinds of rooms, and describes methods of measuring noise and of providing for its insulation from spaces where its presence is undesirable.

The author states that four requirements are necessary for good hearing in any room, and lists them as:

1. The sound, whether speech or music, should have an adequate loudness in all parts of the room.

2. The sound energy should be distributed uniformly in all parts of the room, and the sound reaching the listeners should be free from long-delayed reflections which produce interferences or echoes.

3. The room should contain sound-absorptive materials in such amounts, and of such qualities, as will provide a proper balance between the persistence and the cessation of the articulated components of sound. In other words, the reverberation in the room should be long enough to sustain harmony and impart tonal blending of music, and at the same time it must be short enough to prevent the overlapping and the confusing of the separate sounds of articulated speech.

4. The room should be free from all sources of noise, whether of inside or outside origin.

In order to illustrate the fact that an exceedingly small amount of energy is generated by speech sound, the author uses a most striking example: "It would require all the adult women in the United States speaking fortissimo at the same time to generate a single horsepower of speech energy."

The basis of the decibel scale is explained briefly, and a chart is presented showing some numerical values of this scale with some of the noises of frequent occurrence. Thus, the noise level of speech in theatres varies from 45 to 55 decibels and a noisy ventilating fan may give a level of 75 decibels, while such a fan when under proper control may operate with a noise level of only 15 decibels.

The shape of the room greatly affects its acoustics. The shape may be so fixed that sounds will be distributed or concentrated. High ceilings tend toward delayed reflections of sound, and it is pointed out that only a slight delay may give rise to echoes.

Reverberations of sound in speech and music in auditoriums should be kept small and at a frequency of 512 cycles per second should not exceed 1 to 1.5 sec. under optimal conditions. Reverberation of sound is affected by the material enclosing the room and may be controlled by the use of acoustical absorbent material placed on these surfaces.

The effect of noise on the hearing of speech in an auditorium depends to some extent on the ability of a speaker to raise his voice above the noise, and he attempts to do this provided both he and the audience hear the noise. From this the author observes that:

"If there must be some noise in an auditorium it is much better to have this noise in the proximity of the speaker than in the proximity of the listeners, since the noise will tend to increase the loudness of the speaker's voice, and will not be so bothersome to the audience."

Regarding the matter of tolerable noise levels, the author presents a table of what he calls ideal conditions, and while he notes that these values are seldom obtained in existing buildings, he states that they can be secured by proper design and control, and feels that they represent the limit of noise which should not be exceeded by noise from heating and ventilating equipment.

Studios for the recording of sound, as talking picture studios	**6 to 8 db.**
Radio broadcasting studios	**8 to 10 db.**
Hospitals	**8 to 12 db.**
Music studios	**10 to 15 db.**
Apartments, hotels and homes	**10 to 20 db.**
Theaters, churches, auditoriums, classrooms and libraries	**12 to 24 db.**
Talking picture theaters	**15 to 25 db.**
Private offices	**20 to 30 db.**
Public offices, banking rooms, etc.	**25 to 40 db.**

With such a table the contractor can reduce his problem to fairly simple terms, as the author points out by an example. Suppose that he must guarantee that his equipment will keep noise levels at no higher than 15 db. in a certain theater. If he knows that the noise at entrance to the ducts is at a level of 60 db. he can compute the lowering of level necessary in the duct to be 60-15 = 45 db. He must, therefore, provide enough insulating properties in the duct to absorb this amount.

Three methods of measuring noise are briefly described, each of which involves the use of an instrument. These are the tuning fork, the audiometer, and the acoustimeter. Each may be used to obtain a measure of noise but each presents its own problem

of manipulation and its own limit of accuracy.

Practical methods of reducing noise consist of operating equipment more quietly, of so locating and designing equipment rooms that noises will not be transmitted through the building construction, of sound-isolating rotating and vibrating machines from the building structure, and of preventing sound transmission through ventilating ducts.

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Conductivity of Concrete

By F. C. Houghten and Carl Gutberlet

A.S.H.V.E. paper, annual meeting, Cleveland, January, 1932. Approximately 2500 words, one table and three figures. Appended is a bibliography of seven titles.

A survey of the work of various investigators on the conductivity of concrete shows a wide variation in the reported results. These vary from 2.3 to 17.4 B.t.u. per hr. per sq. ft. per in. thickness per deg. temperature difference. Part can be accounted for by the different mixes of concrete and different methods of testing. As a result of the survey, the authors suggest the use of 12 as the conductivity of concrete until such time as more information is available.

The paper also describes tests made on two concrete blocks over a period of some 1300 days from the date of pouring. These blocks were each 8 in. thick and 2 ft. square. One was of 1:2:5 and the other of 1:2:4½ proportions. They were stored in the laboratory, and were tested from time to time in a guarded hot-plate apparatus to determine their conductivity. When the conductivities are plotted against time, the resulting curves show the rate of change of conductivity for these samples.

In the curves as plotted neither the conductivities nor the rate of change for the two blocks is the same. The tests were so far separated that there is question as to where the curves should be drawn. The change during the first 200 days after pouring is rather rapid and gradually slows as the blocks age. At the end of the period of study the conductivity was still decreasing. The total decrease of conductivity with age is reported as 18.8% and 9.8%, respectively, for the two blocks. The fact that the conductivity decreases with age is borne out by experience.

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Field Studies of Office Building Cooling

By J. H. Walker, S. S. Sanford, and E. P. Wells

A.S.H.V.E. paper, annual meeting, Cleveland, January, 1932. Approximately 7000 words, three photographs, of which two show testing arrangements, three drawings showing construction details, four tables, six graphs showing results.

Describes and gives the results of tests made in the Union Guardian Building, Detroit. These tests were undertaken because the authors felt that some actual results of the performance of a building cooling system would be of interest to designers.

The authors list the factors which affect the cooling and dehumidifying load of a building under two headings, viz: (a) load due to weather, and (b) load due to occupancy. Their studies as reported in the paper are concerned with the load due to weather conditions, involving, especially, the factors of heat transmission through walls and roof, through window glass, and sun radiation through window glass, and were conducted in four phases as follows:

(a) Determination of cooling requirements of the four exposures of the building.

(b) Measurement of the flow of heat through the building wall.

(c) Studies of the temperatures of the wall.

(d) Studies of the overall cooling load of the building.

Each of these phases is described and discussed separately.

Phase (a) was determined by locating four test rooms of identical size and exposure on each of four sides of the building and measuring the temperature and quantity of cooled air required to keep the temperature constant throughout the working day. Fig. 1

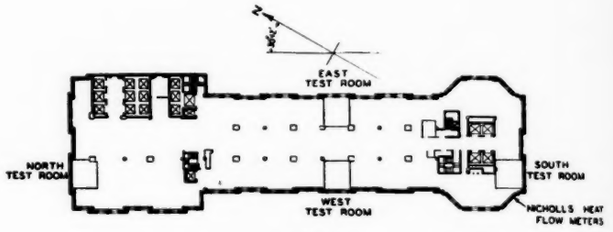

Fig. 1.

shows the location of the test rooms, which were each 14 ft. 8 in. wide and 17 ft. deep. Special arrangements were made to permit measurement and control of the quantity of cooled air flowing through these test rooms. Provisions were also made for temperature measurements.

Readings of the sun radiation were taken with a pyrheliometer located on the top of the south tower of the building. On the days of maximum outdoor temperature these readings were taken at frequent intervals. Observations were also made inside the test rooms to determine the solar radiation entering through the windows.

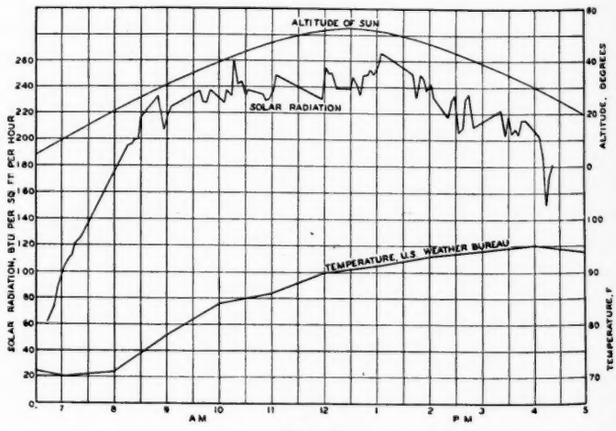

Fig. 2.

Results of the tests made in the four test rooms on September 11, 1931, are given in Fig. 2 and Table 1. This was a hot day with bright sunshine. In Table 1 the *actual cooling load* is that found by the rise in temperature of the cooled air found necessary to circulate through the room. The other items are calculated from the temperature found during the test and using constant flow formula. No allowance is made for infiltration. The heat flow through the wall was found to be too small to plot on Fig. 3, which shows the results graphically. Fig. 3 also brings out the importance of the item of sunshine through glass. In the west room the effect of shading by another building is clearly shown in Fig. 3. The authors state that the results obtained in the four test rooms cannot be considered typical of the total load of the building because of the temperatures maintained, the lack of lights, occupants, and heat from miscellaneous sources, and the differences in the amount of outdoor air circulated, etc.

Phase (b) was studied by means of two Nicholls heat flow meters attached to opposite sides of the south building wall. One meter measured the heat entering the wall from the outside, and the other measured the amount entering the room from the inner side of the wall. Fig. 4 shows how much less heat was received by the room than was received on the outer surface of the wall on certain days when the outside air temperature was high.

Phase (c) was studied by observing the temperatures at the inner and outer surfaces of the walls, and data resulting from temperature measurements made at

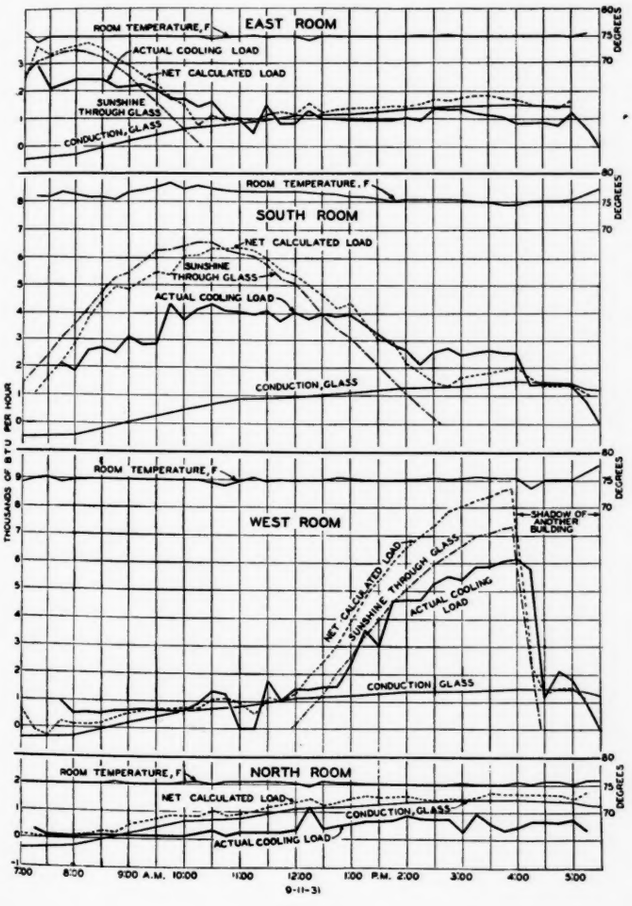

Fig. 3.

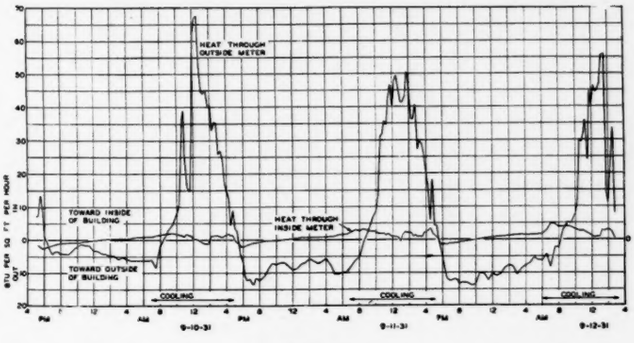

Fig. 4.

various places are also included in the paper. Fig. 5 shows some of this information. At times it was observed that heat was flowing into a wall from both surfaces at the same time; or flowing out of a wall from both surfaces at the same time. One remarkable point in the temperature studies is the small effect which the wide fluctuations in temperature at the outer side of the wall have on the temperature of the room side of the wall. The temperature 1 in. beneath the interior surface varied only 2° during the 24-hr. period of September 11, 1931, while the outside surface temperature varied 33.6°. The temperature of a steel column varied only 1° during this period. In the floor at a point 1 in. beneath the surface, the 24-hr. variation was ½°. At a point 8 in. deep the variation was 1°. All of these data tend to show that the temperature of the interior structure is nearly constant in spite of outside changes, and that if it were not for the disturbance caused by heat entering through the window glass, temperature regulation in the summer would be a simple matter in this building.

Wall conduction was found to be of insignificant magnitude in the total cooling and dehumidifying load.

Fig. 5 shows the relative magnitude of the component parts of the cooling and dehumidifying load *as calculated*[1] for a typical bay having the exposure of the south test room at a time when the sun radiation effect is near its maximum, and assuming 75 dry bulb and 62 wet bulb inside, 85 dry bulb and 72 wet bulb outside, 500 c.f.m. to room, 10% make-up air, 5 people, two 200-watt lamps, and average allowance for fans and pumps. The amount shown for the sun heat through glass is not constant, and the relative magnitudes given in the figure do not apply to all buildings, nor are all the factors considered.

Phase (d) was studied by obtaining the heat removed by the condenser and subtracting the heat equivalent of the power input to the refrigeration compressors. This gave an approximation of the cooling and dehumidifying load. Figs. 6 and 7 show the results of this study.

During the period May to October, inclusive, the refrigerating apparatus operated 1203 hr. Of this time, all three compressors were in operation only 98 hr., or 8% of the time.

The curves of Fig. 6 show a noticeable correlation between the total daily cooling load and the average daily outside temperature.

[1] Italics by HEATING AND VENTILATING.

The following conclusions are listed:

"Sunshine through the window glass is apparently the most important factor to contend with in the cooling of a room. At times it may account for as much as 75% of the total cooling necessary. Because of the importance of the sunshine, cooling systems should be zoned so that the side of the building on which the sun is shining can be controlled separately from the other sides of the building. Consideration should also be given to shadows from nearby buildings which may cover part of the sunny side of a building. If buildings were provided with awnings so that the window glass were shielded from sunshine, the amount of cooling required would be reduced and there would also be less difference in the cooling requirements of different sides of the building.

"In modern office buildings in the latitude of Detroit, the amount of heat entering through the walls in summer is of little importance compared with that coming through the windows. It appears that the use of double windows would reduce the amount entering by direct radiation from the sun as well as that by conduction.

"A study of the cooling for the entire building from Fig. 6 indicates that the total amount of cooling for any day (Mondays and days following holidays excluded) is in general proportional to the difference between the average temperature for the day and 55°.

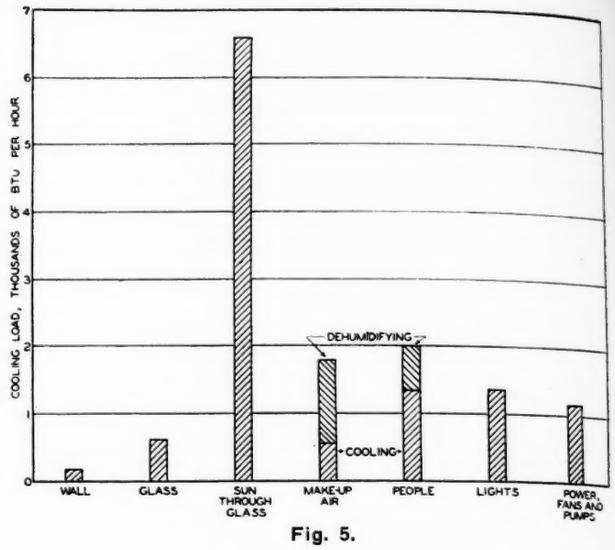

Fig. 5.

It is affected somewhat, however, by the outdoor conditions prevailing during the previous day.

"In the latitude of Detroit, appropriate design figures for the heat entering the walls and windows of an office building are as follows:

Wall transmission—3 B.t.u. per hr. per sq. ft. of wall.

Glass transmission—17 B.t.u. per hr. per sq. ft. of glass.

TABLE 1.

Test Room	Time	Dry-Bulb Temperature, Deg Fahr: Outdoors near Wall	5 Ft from Floor	1 Ft from Floor	2 Ft from Ceiling	Average Outside the Room	Furred Space in Ceiling	Furred Space Beneath Floor	Air Entering	Air Leaving	Volume of Air Entering Room, cfm	Heat Removed from Room by Air, Btu per Hr	Sunshine Through Window Glass: Normal Area, Sq Ft	Btu per Sq Ft per Hr	Btu per Hr	Heat Through Glass by Conduction, Btu per Hr	Heat from Surrounding Space, Btu per Hr	Net Heat Entering Room, Btu per Hr[c]
East	7 A. M.	.....	76.0	75.5	75.3	76.5	74.0	.....	66.9	75.8	.....	.....	30.5	88	2685	-460	-84	2368
	8	.....	75.0	74.5	74.5	75.8	73.5	74.6	64.9	74.5	231	2435	22.0	161	3540	-290	162	3639
	9	.....	75.0	74.8	74.9	75.4	73.0	.....	65.0	74.8	204	2198	12.5	205	2560	210	-68	2929
	10	.....	75.0	74.4	74.5	75.0	73.0	74.6	65.6	74.7	175	1748	2.8	219	615	670	-103	1409
	11	.....	74.9	74.2	74.5	75 0	73.0	.....	64.5	74.8	88	996	0	0	0	850	-29	1048
	12	.....	75.1	75.0	74.9	75.3	73.0	74.8	65.0	75.1	75	832	0	0	0	1110	-161	1176
	1 P. M.	.....	75.0	74.5	74.4	75.3	73 0	.....	64.9	75.0	87	945	0	0	0	1190	-42	1375
	2	.....	75.0	74.8	74.8	75.5	73.0	74.3	65.0	75.0	93	1020	0	0	0	1350	-135	1442
	3	.....	74.9	74.2	74.5	75.5	73 0	.....	65.0	74.7	128	1368	0	0	0	1440	85	1752
	4	.....	75.0	74.8	74.8	75.5	73.0	74.5	64.9	74.9	74	812	0	0	0	1500	-47	1685
	5	.....	74.8	74.3	73.8	75.0	73.0	.....	64.9	74.8	112	1218	0	0	0	1430	21	1678
South	7 A. M.	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	16.0	88	1410	-480	.....	.....
	8	.....	76.5	77.0	76.5	75.3	76.0	76.2	68.0	75.5	229	1885	22.0	161	3540	-430	-544	2800
	9	83[a]	77.0	77.2	77.2	75.4	75.5	.....	66.5	75.5	320	3160	26.6	205	5455	80	-921	4845
	10	87.5	77.4	77.5	77.5	75.7	75.0	75.0	66.0	75.9	348	3780	29.3	219	6415	510	-1084	6075
	11	91	77.0	77.0	77.2	75.7	74.8	.....	64.5	75.4	338	4045	28.2	219	6175	900	-930	6379
	12	95	77.0	77.0	77.0	76.0	74.5	74.5	64.5	75.3	343	4070	22.2	228	5060	950	-856	5388
	1 P. M.	98	76.0	75.8	76.0	76.7	75.0	.....	64.9	75.0	355	3935	13.0	237	3080	1110	-44	4380
	2	98	75.5	75.5	76.0	75.0	75.0	74.2	65.5	74.5	268	2648	4.7	223	1050	1300	-421	2163
	3	98.5	75.4	75.0	75.4	76.0	75.0	.....	66.5	74.5	280	2460	0	0	0	1340	74	1648
	4	97	74.5	74.1	74.8	75.3	75.0	74.5	65.0	73.5	270	2521	0	0	0	1500	307	2041
	5	.....	75.4	75.2	76.0	74.7	75.0	.....	67.2	75.0	159	1360	0	0	0	1460	-308	1386
West	7 A. M.	71.8[b]	74.5	73.8	74.1	76.0	75.0	.....	65.2	74.5	.....	.....	0	0	0	-350	782	658
	8	71.7	74.9	74.2	74.7	75.3	74.2	75.4	66.3	74.8	60	557	0	0	0	-280	210	156
	9	75.7	74.8	74.1	74.4	75.0	74.0	.....	66.2	74.5	76	688	0	0	0	210	156	592
	10	81.8	75.1	74.5	74.8	75.0	74.0	74.8	66.6	74.8	74	665	0	0	0	660	-88	799
	11	84.6	74.8	74.0	74.5	74.8	73.0	.....	62.9	75.0	0	0	0	0	0	800	-119	907
	12	89.0	74.9	73.9	74.5	75.0	73.0	74.0	62.0	74.1	108	1443	0.7	228	160	1100	-174	1312
	1 P. M.	94.3	75.2	74.0	75.2	75.5	72.8	.....	61.7	74.5	167	2365	11.0	237	2605	1200	-116	3915
	2	96.5	75.0	73.8	75.0	75.3	72.0	74.0	60.5	73.8	313	4615	22.7	223	5060	1320	-77	6529
	3	97.9	75.3	74.5	75.2	75.8	72.0	.....	61.0	74.0	373	5375	33.0	200	6600	1360	-181	8005
	4	99.7	75.4	74.5	75.1	75.8	71.8	74.0	59.0	73.8	373	6120	31.0	183	5675	1440	-251	6868
	5	94.4	74.8	73.8	74.5	75.0	72.0	.....	62.1	74.1	132	1775	0	0	0	1400	-128	1469
North	7 A. M.	[a]	75.0	75.0	75.0	76.0	75.5	.....	74.0	75.5	.....	.....	0	.....	0	-370	322	185
	8	.....	75.0	75.0	75.0	75.5	75.5	.....	73.0	75.0	54	116	0	.....	0	-300	251	184
	9	77	75.0	75.0	75.0	75.3	75.5	.....	72.5	75.0	54	145	0	.....	0	210	108	551
	10	81	75.5	75.0	75.0	75.3	75.5	.....	72.5	75.0	54	145	0	.....	0	680	-34	879
	11	84	75.5	75.0	75.0	75.3	75.5	.....	72.5	75.0	94	255	0	.....	0	760	-34	959
	12	87	75.2	75.0	75.2	75.3	75.5	.....	72.2	75.0	111	335	0	.....	0	1100	36	1369
	1 P. M.	88	75.2	75.0	75.0	75.6	75.0	.....	71.2	75.0	132	545	0	.....	0	1190	71	1494
	2	90	75.0	75.0	75.0	75.0	75.0	.....	69.5	75.0	139	834	0	.....	0	1340	0	1573
	3	92	75.2	75.0	75.0	74.5	75.0	.....	70.0	75.0	48	261	0	.....	0	1420	-200	1453
	4	94	75.5	75.0	75.0	75.5	75.0	.....	70.5	75.5	75	410	0	.....	0	1430	0	1663
	5	.....	75.0	75.0	75.0	75.5	75.0	.....	70.5	75.0	146	714	0	.....	0	1360	-143	1450

[a] Temperature taken 6 in. from wall. [b] Temperature taken 1 in. from wall. [c] Heat given off by observers assumed to be 100 Btu per hour. Heat entering through wall assumed approximately 125 Btu per hour in east and west rooms and 135 Btu per hour in north and south rooms.

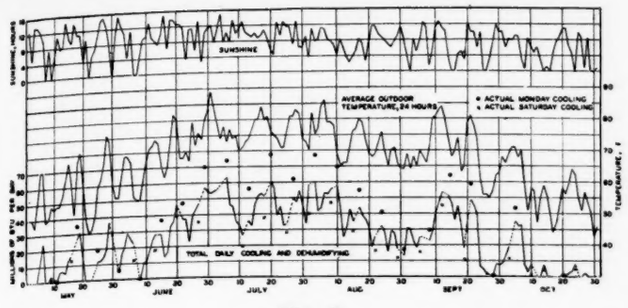

Fig. 6.

Sunshine (east and west sides)—160 B.t.u. per hr. per sq. ft. of glass.

Sunshine (south side)—140 B.t.u. per hr. per sq. ft. of glass.

"The maximum rate at which solar radiation enters

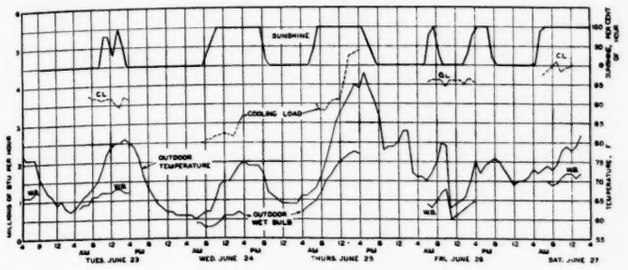

Fig. 7.

south windows is less than for east and west windows because of the angle at which the sunshine passes through the glass. The glass area referred to is the area actually exposed to the sunshine. If there are overhanging ledges above the windows, these must be taken into account."

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Transmission of Radiant Energy through Glass

By R. A. Miller and L. V. Black

A.S.H.V.E. paper, annual meeting, Cleveland, January, 1932. Approximately 3000 words, two photographs of testing apparatus, seven curves and graphs, principally showing test data.

Describes test methods used to determine the amount of radiant energy transmitted through glass. Glass test specimens were inserted between a source of radiant energy and a pyrometer. The source of energy was kept at a constant temperature and the record taken shows how the radiant energy absorbed by the pyrometer varies. The glass itself appeared to pass no radiant energy, but when it became heated the heat passed through by conduction, and some of this reappeared on the other side in the form of radiant heat.

The following conclusions are included:

There is practically no loss of energy from a building by direct transmission of radiant energy through the windows. Some energy is lost by the glass absorbing the radiant energy, and thereby being heated and then radiated to the outside. This loss, however, is believed to be insignificant compared to the losses due to convection currents and conduction. Two plates of glass separated by an air space cut off the greater part of the radiation loss and reduce the conduction loss to a comparatively low figure.

It seems evident that the increasing use of glazed areas in buildings need not mean increased heat losses if multiple panes of glass, separated by air spaces, are used.

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Properties of Metal Foil as an Insulating Material

By J. L. Gregg

A.S.R.E. paper, annual meeting, Cleveland, January, 1932. Approximately 3000 words, four tables, and three charts and graphs.

Reviews the experimental work of others on thin metal insulators and describes the experimental work and results of the Battelle Institute. While the title states that metal foil was considered, the paper refers constantly to aluminum foil only.

Following a general statement of the considerations which might make the use of metal foil insulators desirable commercially, the author briefly reviews the theory which is usually applied to heat transfer through thin foil insulation formed by flat plates spaced parallel one to another. He notes that the convection transfer will vary with the width of the air space.

In the experimental work, a hot-plate method was used to determine conductivity and many methods of using foil in insulating units were studied.

Most of the units were formed from plates made by mounting bright aluminum foil on both sides of paper or cardboard. It was found, however, that substantial units could be made from unmounted foil. Many types of separators were tried, and it was found that even though only a small percentage of the space was occupied by the separators, the use of a material of relatively high thermal conductivity apparently increased the conductivity of the unit by an appreciable amount. With respect to conductivity, wooden separators are much more satisfactory than cardboard separators because of the lower heat conductivity of wood.

From values given in tables accompanying the paper, it is concluded that effective insulators can be produced by spacing foil to form ½ in. air layers. The weight of the insulators can be reduced by using unmounted foil, and possibly by using thinner separators. A plotted curve presented with the paper shows a sharp increase in conductivity when the width of the air space exceeds ½ in. This leads to the thought that there is an absence of convection when the air space between layers of foil is ½ in. or less. Additional evidence of the absence of convection with a ½ in. air space was found when the sample was mounted horizontally and heat was applied first to the top and then to the bottom. As no difference of conductivity was found, it seemed that no convected heat was being transferred under these conditions.

An attempt was also made to determine the permanency of the high reflectivity of aluminum. Results so far obtained indicate that except for anodically-

treated aluminum, the emissivity is not especially increased until the aluminum is covered by a visible layer of corrosion products.

Values for the conductivity of the insulations of foil studied are reported as ranging from 0.22 to 0.61 B.t.u. per hr. per sq. ft. per deg. F. per in. thickness depending on the construction used. Values of no greater than 0.25 were reported with a number of the constructions used in the studies. Such values are lower than those usually reported for the solid materials used as insulators.

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A Study of Intermittent Operation of Oil Burners

By L. E. Seeley and J. H. Powers

A.S.H.V.E. paper, annual meeting, Cleveland, January, 1932. Approximately 3000 words, 11 curves showing results of the study. Report of joint research conducted at Yale University by the A.S.H.V.E. and American Oil Burner Association.

Presents the results of laboratory studies, particularly as regards economy or efficiency of operation. Explains briefly the ideas, methods and necessity for operating oil burners intermittently. The authors divide operating methods into three kinds, viz: on and off, high and low, and continuous. The tests were conducted using the A.S.H.V.E. proposed code.

In some of the tests reported, the on period was arbitrarily set at 30 min., and the off period was varied to obtain the rate of burning desired. In these cases the resulting efficiency curves showed the same general shape. As the setting of the on period at 30 min. left some question as to what would happen if it were shorter, additional tests were made in which the length of the period was varied. The efficiency curves, however, continued to show the same characteristics.

The author also discusses the influence of boiler characteristics on the efficiency, the effect of furnace draft, and the setting of the burners for capacity.

Following conclusions are listed:

1. The intermittent efficiency curve for *on and off* and its equivalent for other types of burners is the true indicator of fuel economy to be expected in actual practice.

2. All intermittent curves or equivalent have substantially the same characteristic shape for similarly adjusted burners.

3. Any number of boiler and burner combinations which have the same capacity, efficiency, furnace draft and excess air or combustion setting while *on* will produce substantially identical results in actual use.

4. Continuous-type burners are influenced by the heat absorption efficiency of the boiler over the whole range of output.

5. *High and low* burners tend to follow the *on and off* type of efficiency curve. The curve should be confined between low and high limits of efficiency actually obtained by the *low* and the *high* flames, respectively.

6. The lower the furnace draft setting and the more uniformly it is maintained, all other things being equal, the better the fuel economy.

7. Various *on* period timings seem to have practically no effect on the intermittent efficiencies.

8. The best economy on intermittent operation for any combination is an *on* setting which produces the highest efficiency.

9. It is possible that certain types of heating systems may be better adapted for economical intermittent operation than others, but this is not definitely known.

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A Study of the Combustible Nature of Solid Fuels

By R. V. Frost

A.S.H.V.E. paper, annual meeting, Cleveland, January, 1932. Approximately 2000 words, 12 figures.

A partial report of studies undertaken by the author to learn the reasons for the variations encountered in the burning characteristics of coals, and to devise methods of correction. The paper consists largely of a presentation of some of the results obtained when the relationship between draft and the rate of combustion and the relationship between rate of horizontal fire travel and rate of combustion were studied during the course of the investigation.

Coals are roughly classified as to their burning characteristics by calling them either hard or free-burning. Previous to the author's studies there were no tests to determine into which class given coals fall. The draft required to maintain a given rate of combustion is considered as a test which throws some light on the subject. The author's curves show that the draft required varies greatly. He shows that by blending he was able to produce coals which were much less variable in their burning characteristics.

Rate of horizontal fire travel is taken to be an index of the rapidity with which fire will spread from one part of the fuel bed to another. Its usefulness lies in the fact that a coal which spreads the fire quickly is a free-burning coal. The author's curves show how variable this feature is in different coals.

The following conclusions are listed:

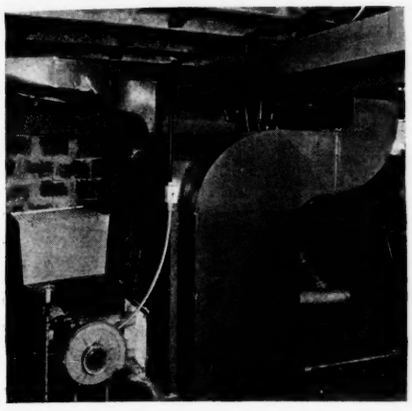
It is apparent that the usual tests of coal fail when applied to the determination of the combustible character of the coal. Neither appearance, density, chemical analysis nor location can be used as an indicator. The only positive means of determining the combustible character of a coal before it is put to use is to make a specific test for that particular purpose.

That the determination of the combustible character of the coal is highly essential to its proper utilization cannot be denied. Under hand-firing practice it was possible to blunder along with efficiencies of 50% or 60% without serious complaint, but with the increasing employment of automatic burning devices and the increasing use of coke, oil and gas as competitive fuels, a coal that is not uniform in combustible character becomes a drag upon the entire practice of coal utilization, for with the absolute necessity of close adjustment and balance between air and coal feed in the automatic burning of coal, any coal that is not uniform in combustibility is immediately condemned.

New Equipment

Lansing Dailaire Warm-Air Systems

Dail Steel Products Co., Lansing, Mich., is marketing the Lansing Dailaire System, a domestic air conditioning unit. The furnace is designed to burn either gas or oil, the combustion and heating chamber being constructed of thin chrome alloy steel plates, which are corrosion and scale resistant. The combustion chamber is wedge-shaped and carries the warmed air rapidly upwards, and 12 large flues furnish the radiating surface.



Dailaire warm-air installation

A rock wool insulating blanket is placed between the primary and secondary heating surfaces to prevent reheating of the gases by direct contact with surfaces exposed to the radiant fire.

Circulation is induced by means of a blower at the top, enclosed in the furnace casing. Cold air from the rooms is drawn into the air washers located on each side of the casing where dust, smoke and odors are removed by a fine spray of water, which also serves to humidify the air. The sprays do not operate except when the blower is working, and the blower is thermostatically controlled to prevent cold air from being blown into the rooms.

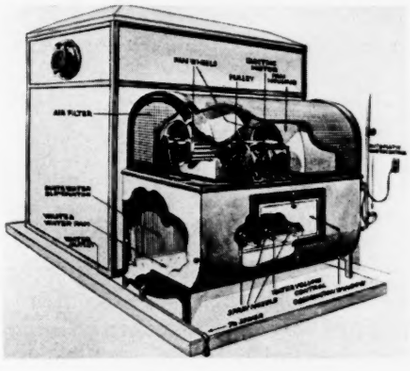
By operating the blower and air washer alone in warm weather, an effective cooling system is provided, which will insure circulation of the air in the rooms and reduce the temperature.

The furnace is supplied in four sizes, the output ranging from 85,000 to 300,000 B.t.u. per hr., to supply leader pipe areas ranging from 625 sq. in. to 2200 sq. in., when operated on gravity. When used with the blower, the output ranges from 115,000 to 560,000 B.t.u. per hr.

Holland Air Conditioning Unit

Announcement has been made by the Holland Furnace Co., Holland, Mich., of a moderate-priced domestic air conditioning unit. The conditioner is an auxiliary unit in the form of a compact sheet metal cabinet for attachment immediately to the rear of the casing of a warm-air heater. It occupies less than 10 sq. ft. of space and is about 40 in. high. In addition to the cabinet itself, the conditioner consists of four principal parts; a semi-cylindrical copper mesh filter over the top of the cabinet; a high-speed, twin squirrel cage type fan unit with a ¼ h.p. motor between the fans and, within the cabinet a series of spray nozzles that emit fine, swirling, water sprays, and a reversible eliminator.

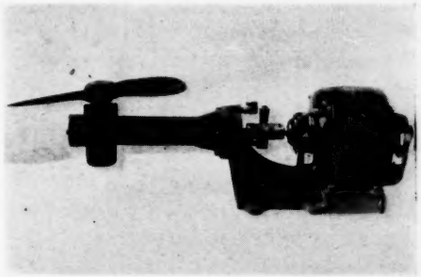
The fans, with capacity for moving 1500 to 2000 cu. ft. of air per min., draw the home air through the filter and force it through the water sprays. The water sprays impinge against the eliminator which is of "Z" plate construction, the water dripping off into the drip-pan bottom of the unit and draining into the sewer, carrying with it dust and other impurities removed from the air. The washed air then passes on through the eliminator into the heating plenum of the heater and thence is forced into the rooms of the house.



Sectional view of Holland air conditioner

Motion Picture Booth Exhaust Fan

The Davenport Mfg. Co., Meadville, Pa., is marketing a propeller type exhaust fan designed for use in motion picture projection booths.



Davenport exhaust fan

The fans have a capacity of 500 c.f.m. to 800 c.f.m., and are available in several types depending on the arrangement of the exhaust duct.

Frigidaire Room Coolers

Frigidaire Corp., Dayton, Ohio, showed for the first time at the heating and ventilating exposition in Cleveland its line of room coolers.

The line consists of three models. The first (H type), similar in size to the conventional radiator of modern design, consists of a cooling coil containing a refrigerant, fans, moisture controlling features and a heating coil to be connected with the hot water or steam heating plant. A second, taller unit (V type), comparable in size and appearance to a highboy radio cabinet is provided. A smaller, cube shaped unit (S type) is designed for installations where floor space is at a premium. It may be suspended from the ceiling or wall or the equipment may be recessed in the wall.

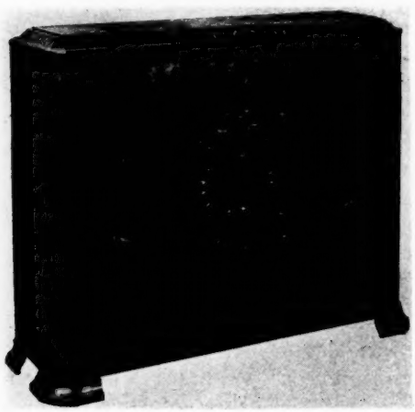
In most instances, the compressor unit will be placed in the basement or an adjacent closet. For offices and other rooms in buildings where such installations are impractical, a sound-proof, polished desk-size cabinet will conceal the compressor.

The conditioning unit and the

compressor are connected through a conduit. Joining of the heating coil with the existing heating system may be done by any contractor. All mechanism is concealed.

The H and V models are equipped with centrifugal fans powered by 1/20 hp. motors. The air movement in the V models is at the rate of 450 c.f.m., in the H models at 550 c.f.m., and in the S model, which has a bladed fan instead of a centrifugal fan, is at the rate of 585 c.f.m.

A fan switch and compressor switch, mounted in the end of the cabinet, control the V and H models. The S model is governed by two similar switches mounted separately. Thermostats are optional.



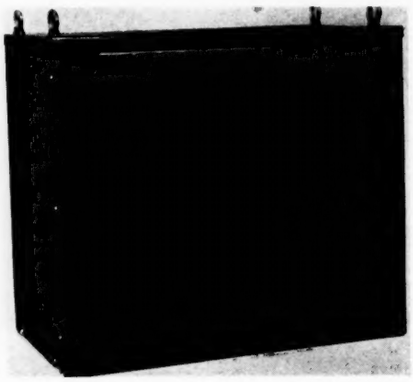
H type cooler

The sound-proof housing for installations where the compressor is placed in the same room with the conditioner is available in burled walnut wood. Oak or mahogany wood is available on special order. Its overall dimensions are 46 in. in width, 30 in. in depth and 30 in. in height. This housing may be used in remote installations if desired. The foundation is a 3 in. thickness of corkboard, and the walls and top are sound-proofed by 2½ in. layers of insulating material.

Frigidaire 1½ hp. compressors are used for individual installations. The compressors are available for all cycles and voltages of current.

Copper tubes carry the refrigerant to the conditioner from the compressor and return it. A small pipe for draining of the dehumidification overflow pan and the refrigerant tubes are concealed in a conduit.

In the V and H models, room air is drawn from the floor and into the conditioner by a centrifugal fan and blown over the cooling coil, which is maintained at an average temperature of 45°. In its passage over the cooling coil, the air is chilled and excess humidity removed. The percentage of dehumidification is dependent upon the relative humidity and the temperature of the room.



S type cooler

After the air is cooled it is sent through a 20° grille on the top of the conditioner cabinet and diffused throughout the room with only a slight draft. Cooling begins instantly when the compressor is started and the air is moved at a rate that provides a complete cycle within 8 min. in the average room.

In the colder seasons, when cooling and dehumidifying are unnecessary and heating is needed, the refrigeration unit becomes inoperative and the heating coil, connected to the owner's existing hot water, vapor or steam heating system, begins to function. The heated air is moistened by dissipation of water from the humidifier, and is sent into the room by the centrifugal fan. Water for humidificaton is obtained by tapping of the regular water system and is carried to the conditioner by a small copper tube concealed in the conduit.

Kleenaire Filters

An all-metal filter for use in warm-air heating, is being manufactured by the Kleenaire Filter Co., Stevens Point, Wis. The device makes use of corrugated woven wire cloth arranged to form zig-zag channels in the filtering media. The dust-laden air is cleansed by centrifugal action in following its course through the filter, as well as by the affinity of dust for dust.

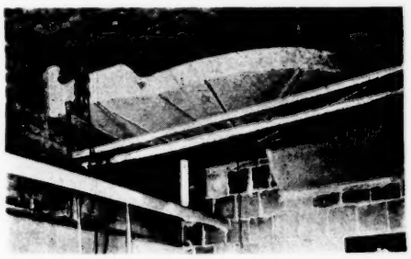
The filters, which can be used dry for the lower velocities and with viscous substances for higher velocities, are available in two sizes, the surface areas being the same in both cases. These are 20 in. x 20 in., and 16 in. x 25 in. Units are made in two types of construction, recommended for use dry for velocities not exceeding 300 to 350 ft. per min., and for velocities not exceeding 200 ft. per min. Another type is made for use in gravity warm-air systems.

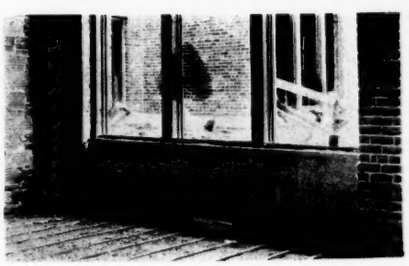
Light rapping cleans the filters when used dry, and flushing with hot water and soda, when used with viscous substances at high velocities.

Minfelt Insulating Cement

Minfelt, an insulating cement which will stick to any clean metal surface, hot or cold, is being manufactured by The Mineral Felt Insulating Co., Toledo, Ohio. It is recommended for application to ducts, radiator recesses, and similar uses.

In addition to its adhesive qualities, the material is extremely resilient, and is not loosened by expansion, contraction, or vibration. It is mixed with fresh water to proper consistency, and a rough coat ½ in. thick is applied to the





(Above) Minfelt insulation on warm-air duct. (Below) on radiator recess

cleaned surface. When this is dry, the finishing coat is put in place. Reinforcements are not necessary for thicknesses up to 1½ in.

The material is shipped dry in 50 lb. bags, and 100 lb. of it will cover approximately 50 sq. ft. with insulation 1 in. thick. It is suitable for temperatures up to 1650°.

Century Model D Oil Burner

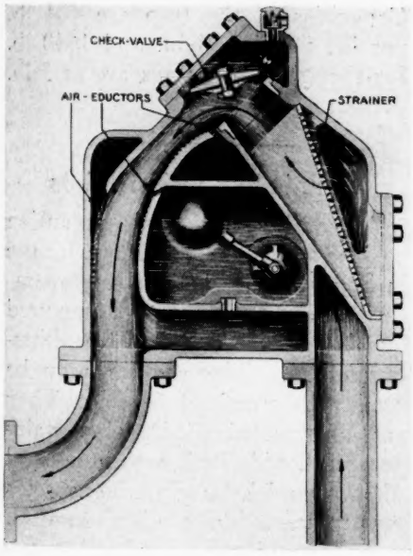
The Century Engineering Corp., Cedar Rapids, Ia., has placed on the market its Model D oil burner with a so-called floating flame. The burner was shown for the first time at the Second International Heating and Ventilating Exposition at Cleveland.

The burner may be equipped with either gas or electric ignition, and has been approved by the Underwriters' Laboratories. The burner is made in two models, one with a capacity of 4 gal. per hr., and the other 12 gal. per hr.

Suction Line Primer

A new priming device for centrifugal pumps has been announced by Barrett, Haentjens & Company, Hazleton, Pa.

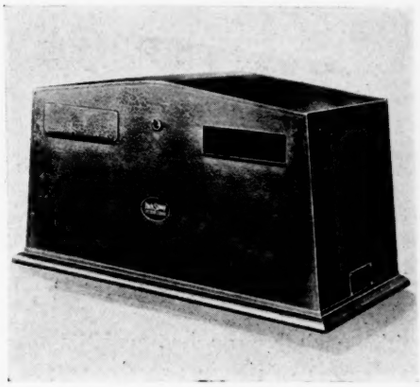
The device combines in one casting a check valve, strainer, and air removing device. It is built in various sizes, for a maximum capacity of 1500 g.p.m. and a maximum operating head of 200 ft. When furnished with a special control panel, it makes possible automatic operation of the pump with protection against loss of water, air leaks, break of the column line, etc.



Centrifugal pump primer

ParkSpray Humidifier

Parks-Cramer Co., Fitchburg, Mass., has introduced the ParkSpray Mistyfier, a compact mechanical humidifier for offices, stores, laboratories, and similar applications.



ParkSpray Mistyfier

Water obtained through a permanent connection to the city water supply is broken up into a fine spray by a high-speed centrifugal humidifier. An automatic humidity regulator starts and stops the device according to the requirements. Circulation is provided by a fan driven by a 110 volt, 60 cycle motor.

Dirt washed from the air in the humidifier is disposed of by removing the mechanical unit from the cabinet, and flushing out the interior.

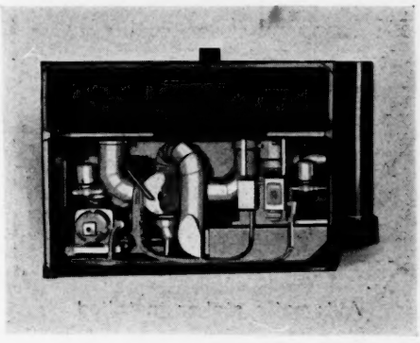
Solenoid Valve for Air Conditioning Apparatus

The Automatic Products Co., Milwaukee, Wis., has announced the Model 71 solenoid water valve, designed particularly for control of water supply on air conditioning refrigeration, and similar uses. The device is 4 in. high, and has a diameter of 2 in. It has a capacity of 75 gal. per min. at a pressure of 45 lb., the port opening being 3/32 in. and the pipe connection ⅜ in. The standard unit is intended to draw 15 watts per hr., at 110 volts, 60 cycle, a.c., but can be furnished for other voltages and frequencies. It can be had with tapping for BX or conduit or with plug-in attachment.

Composition of the valve and seat is designed to eliminate trouble due to leakage, such as stoppage of the spray nozzles on air conditioning apparatus. It is stated that the valve will hold tight on pressures as low as 5 lb., and will open against a pressure as high as 100 lb.

Silica Gel Dehumidifying Unit

A dehumidifier, for use in connection with residential air conditioning, built by the Silica Gel Corp., Baltimore, Md., was exhibited at the recent exposition at Cleveland. It can be used in conjunction with a coal, oil, or gas-fired boiler or furnace, and constitutes a link between refrigeration equipment, which will reduce the dry-bulb temperature, and the commonly used domestic heating systems.



Dehumidifying unit

Silica gel, the dehumidifying agent, is a hygroscopic substance, extremely porous in structure. These pores absorb the moisture from air passed through it. When the gel becomes saturated, it can be reactivated by evaporating the moisture from it by the application of heat. This cycle of absorption and reactivation is automatic.

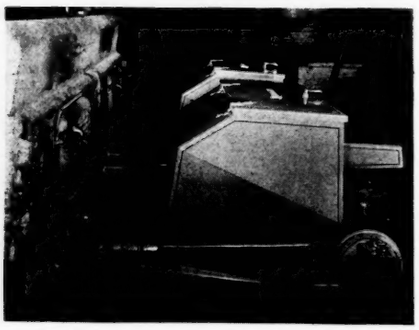
Outside air is drawn into the unit through a filter and is delivered to the beds of gel in the upper part of the apparatus. From here it is passed, in the dehydrated condition, into the cooler, and then to the main circulating system which discharges it into the conditioned areas. The ratio of recirculated air, from the conditioned areas, to the dried air from the outside source can be regulated to obtain the desired results.

Development of the system resulted from investigation by the

Silica Gel Corporation, and from experimental work carried on by the company in cooperation with the Committee of Industrial Gas Research of the American Gas Association. The system has had application in offices, public buildings, and homes.

Great Western Automatic Stoker

Announcement has been made by the Automatic Stoker Corp., Peru, Ind., of the Great Western automatic stoker, using the screw underfeed principle. The device is available in a range of sizes to suit various domestic and industrial requirements. An automatic crusher which



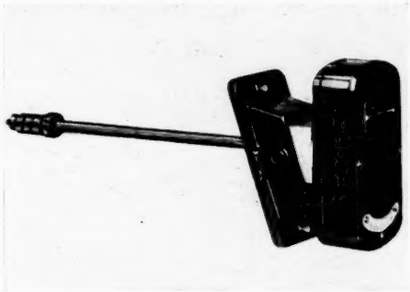
Automatic stoker installed

breaks up the larger lumps makes possible the use of the cheaper grades of coal.

Barcol Limit Control for Warm-Air Furnaces

A three-terminal furnace limit switch, for use in conjunction with a low voltage regulator on warm-air installations, has been developed by Barber-Colman Co., Rockford, Ill. The switch may be used alone as an operating control, or may be used in connection with a standard room-type thermostat as a cutout for excessive temperatures. When used as a cutout, the device is a reliable safety control, protecting the furnace and conserving fuel.

The instrument consists of an adjustable switching mechanism in a phenolic resin box, to the back of which is fastened a long tube carrying a bimetallic helix. Within this tube is a rod which links the free end of the helix with the contact tongue of the switching mechanism. Detent magnets are employed on either side of the contact tongue, eliminating operating errors due to vibration, and making a neutral switching position impossible.



Warm-air furnace limit control

The switch may be mounted on the bonnet of the furnace by means of the flange and stud on the back of the switch case. The flange is capable of holding the switch in a horizontal position on the furnace bonnet. A set screw in the flange holds the switch so that the actuating helix is directly in the path of the warm air.

The device may be adjusted to change at any limiting temperature between 100° and 400°. The construction of the switch provides that a drop in temperature of 60° must take place in the bonnet after the switch has changed from the limiting contact, before the switch will again return the control to the limiting circuit. This differential is constant for all temperature limits at which the switch will operate.

Duo-Vitol Air Conditioner

A new product of the General Regulator Corp., Chicago, Ill., is the Duo-Vitol air conditioner, for use in clubs, hotels, homes and institutions. Electrically-driven, the apparatus is entirely enclosed in an American walnut cabinet.

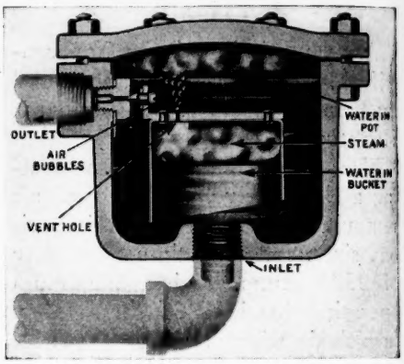
Air is drawn in at the bottom and is discharged in three directions from the grille at the top of the device. The conditioning process consists of purification, deodorization, charging with ozone, and ultra-violet ray solarization, in addition to the washing of the air and humidification control. The degree of humidification can be regulated, and the driving motor is of the varying speed type, permitting of regulation to meet all demands.

It is claimed that a single unit will condition and circulate the air in an apartment or small home. The apparatus can be rendered portable for hospital and institution use by applying a roller carriage.

Crane Inverted Open Float Steam Trap

Crane Co., Chicago, Ill., has announced an open float steam trap in ½ in. and ¾ in. sizes for pressures up to 150 lb. Intended for use with small steam heated units such as heating coils, unit heaters, steam mains and risers, it can be used on both low pressure and vacuum systems.

The float, which is drawn in one piece and has a needle-point valve mounted on the top, is fitted with a small vent hole for the release of air and other non-condensable gases. With the inlet connected at the bottom center of the body the discharge can be turned to any desired angle.



Sectional view of steam trap

Capacities of the trap are 404 lb. per hr. for the ½ in. and 1640 lb. for the ¾ in. Ratings are at 5 lb. pressure.

Dennis Expansion Joints

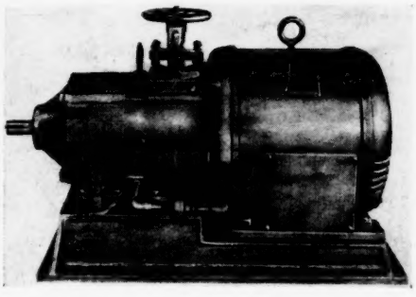
A new line of Dennis expansion gaskets and expansion joints, for use in high pressure and high temperature piping, is being manufactured by Thomas R. Heyward Co., Pittsburgh, Pa. The gaskets may be used with standard flanges. They are constructed of Duraloy stainless steel, Udylited vanadium steel springs, and have graphite-asbestos gasket faces. They will allow axial, lateral, angular, or torsional movement of the connected parts, and

may be used for pressures up to and including 400 lb. pressure, and temperatures up to 750°. By using these gaskets in flanged swing joints, it is said that practically unlimited lengths of expansion can be taken care of without strain in the lines.

Stainless steel is also used for the bellows. Forged steel flanges are attached by cast recessed welding on expanded tube ends. Flexible reenforcement of the bellows is accomplished by the use of flexible bearing plates and longitudinal web strips.

Electro-Hydraulic Speed Transmission

American Engineering Co., Philadelphia, Pa., has placed on the market a variable speed transmission which, when connected to a fixed speed prime mover at one end, will deliver any speed from zero up to the capacity of the unit at the other end. At the present time 5 hp. and 15 hp. units are available, with 3



Electro-hydraulic speed transmission

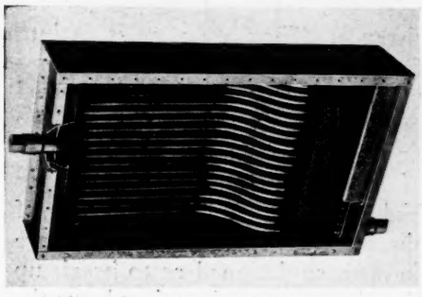
hp. and 10 hp. units being developed. Present uses include applications to stokers, refrigeration units and compressors, etc.

Flexitube Aerofin Heating Surface

Announcement has been made by the Aerofin Corp., Newark, N. J., of the Flexitube Aerofin fan system heat surface, suitable for intermediate pressures to 200 lb.

Tubes are of the usual Aerofin construction, having the helical extended surfaces wound on seamless copper tubing. These are placed in the casing so that opposite header joints are not in line, the tubes being offset. This bending allows the tubes to expand or contract under temperature changes.

Casing dimensions are as in previous construction. Steam supply connections are on the center-line of the casing, with respect to width, and the returns are eccentric on the



Heating surface with flexible tubes

opposite end. Both are standard male pipe thread.

Burgess-Parr Duplex Pressure Regulating Valve

Burgess-Parr Co., Division of C. F. Burgess Laboratories, Inc., 111 W. Monroe St., Chicago, Ill., has placed on the market the Burgess-Parr duplex pressure regulating valve, designed for use on high pressure oil burning systems.

The functions of the device on an oil burner are: It admits oil to the burner nozzle only when an atomizing pressure has been reached; it maintains a definite constant pressure at the nozzle; it shuts off abruptly when the pump stops, thus preventing oil accumulation in the furnace; it prevents oil siphoning when the tank is above the burner, and it by-passes oil to the tank when the fuel is in excess of the amount required for combustion.

The device consists of two cylinders in each of which is a stainless steel piston-operated needle valve. Oil enters the valve and exerts force against both pistons, until at a specified minimum pressure the burner valve is lifted, and at a specified maximum pressure the by-pass valve is lifted, thus maintaining the pressure to the burner between these prescribed limits. Cadmium-plated compression springs above the pistons provide a force against which the oil pressure must work, and adjusting screws are provided to regulate the spring pressure, and hence the oil pressures at which the valve will open and close. When the pressure reaches a definite amount, one piston is lifted, permitting oil to flow to the nozzle. At a higher pressure the second piston is lifted, and the excess oil flows through the by-pass, thus limiting the maximum pressure of the oil delivered to the nozzle. To insure seating of the needle, and, consequently, sharp cut-off of the oil at the nozzle, a special floating construction is employed for the minimum pressure valve.

Reeves Vertical Speed Transmission

The Reeves Pulley Co., Columbus, Ind., has announced the Reeves vertical enclosed design variable

Vertical speed transmission

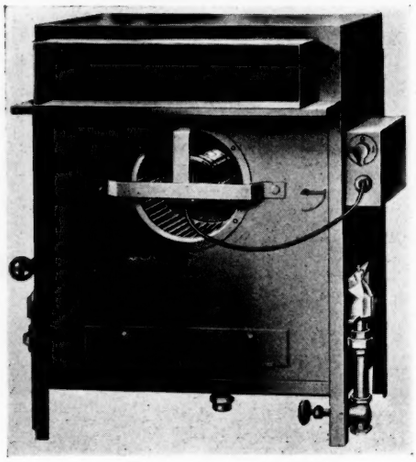
speed transmission, which is available in 10 sizes covering speed ratios from 2:1 to 8:1, inclusive.

This unit requires less floor space than the horizontal type. An outstanding feature of the design is the convenience in lubrication facilities. It is necessary to lubricate only once every 30 to 45 days, depending on the type of service.

Zephyr Air Conditioner

Air-control Systems, Inc., Chicago, Ill., recently announced the Zephyr Air, an air conditioner for supplying washed and humidified air to offices and homes, and for air cooling in warm weather. The apparatus may be placed in the cellar, in a closet or in the living quarters. When located out of sight, no cabinet is necessary, but for use in

New Equipment



Front view of Zephyr Air

office or living rooms an attractive enclosure is provided.

Connection is made to the steam or hot water heating system. The air is circulated through a copper-fin type radiator. The heated air is then subjected to washing and humidifying by means of a fine spray of water, after which it is discharged back into the room. When used for cooling, refrigerated water supplied by a one-ton machine, in remote location, is used in place of the steam or hot water.

It is said that the device will replace one ordinary radiator, and will supply humidification for 40,000 cu. ft. As a cooler it will serve from 4000 to 6000 cu. ft.

Uni-flo Grilles

A line of grilles and registers for forced air heating, ventilating, air conditioning, concealed radiators, etc., has been placed on the market by the Uni-flo Grille Products Co., 2265 W. Grand Blvd., Detroit, Mich. According to the manufacturer, the reduction of air flow through the grille, rather than the free air area, was the important factor in the design of the line of grilles. It is claimed that the company's grilles more completely diffuse the air stream as it flows into the room.

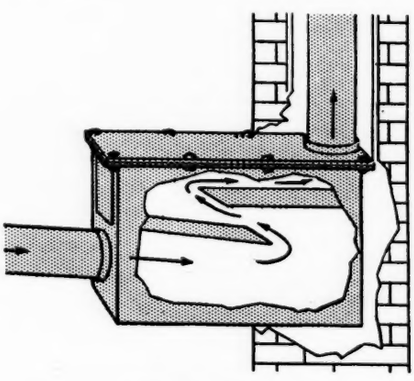
The cores of the grilles are made of a series of hard, cold rolled nickel finish steel fins, 0.020 in. thick, spaced ¼ in. apart horizontally and supported by vertical bars spaced 1 in. apart. These supporting bars are made of cold drawn, round edge wire, 1/16 in. thick by ¼ in. wide, the narrow edge facing the front to offer the least amount of air resistance. They are a press fit into flange slots in the fins. The complete assembly is then immersed into a non-ferrous metal bath. The fins are ½ in. deep for the purpose of straightening the air stream at high velocity. In the process of shaping the fins, diffusers are formed, so designed as to break up the air currents when they leave the grille fins.

The grilles are available with and without dampers in standard sizes and shapes.

Condensation Box for Gas Appliances

Announcement of a condensation box which prevents damage by corrosion resulting from condensation in gas appliance flues and chimneys, has been made by the Condensation Engineering Corp., 122 S. Michigan Ave., Chicago, Ill. The device prevents back-draft and up-draft in the chimney from interfering with the operation of the burner, but will not permit cold air to enter the flue and chimney. The box is made of Armco iron covered with vitreous enamel, so as to withstand corrosion.

The upper baffle directs any back-draft out the door above the breeching, which opens outward and will close immediately the back draft has subsided, preventing the drawing in of cold air. A small door in the side of the box is adjustable to permit control of draft conditions. This maintenance of a high temperature within the box prevents condensation while the appliance is in operation, and evaporates condensate formed in starting up.



Condensation box

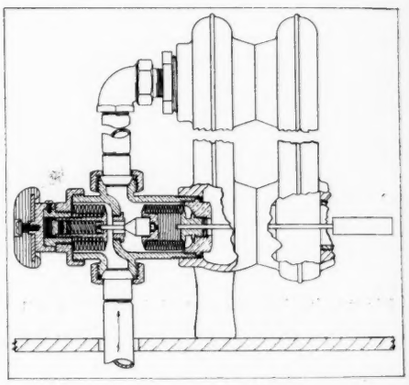
National Gas Boiler

The National Radiator Corp., Johnstown, Pa., has announced the National gas boiler. Steam and hot water boilers are available in ten sizes. Steam capacities range from 560 to 2250 sq. ft. and hot water from 900 to 3600 sq. ft.

The chassis and controls are insulated and incased in a cover designed by Walter Teague. The cover is finished in French grey baked enamel. The base, dome and instrument panel are chromium plated, and may be easily cleaned with a cloth. Deep internal tube sections provide large steam liberating area and steam storage space.

Sarco Selvac Selective Vacuum System

Sarco Co., Inc., New York, has developed a vacuum system with



Radiator control element

which a high vacuum is carried in the returns and in the radiators, the vacuum being varied automatically as the heating requirements change. The system, which has been named the Sarco Selvac selective vacuum system, is designed to maintain a required temperature either in the whole building or different temperatures in different zones.

An essential part of the system is the radiator control element shown in the accompanying illustration, made in various types to suit different kinds of radiation. No trap is used on the radiator outlet, so that the high vacuum carried on the returns is transmitted into the radiator. The Selvac element admits

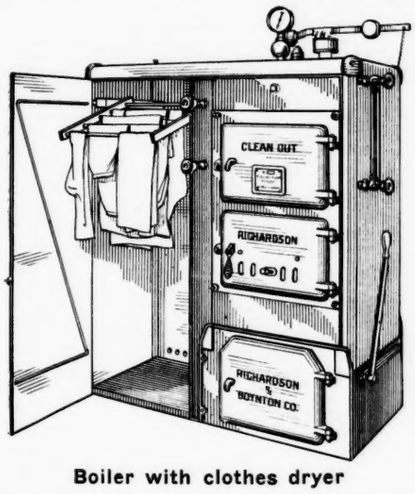
only sufficient steam to maintain the temperature corresponding to this vacuum. Thus, the radiator is filled with steam varying in temperature from 125° to 212° or more.

The vacuum carried on the system is produced by a special vacuum pump. It is regulated automatically in accordance with the outdoor temperature and the temperature desired inside the building by an automatic vacuum control valve. Air pressure from an outside reservoir operates a large diaphragm to open the valve and put the pump vacuum on the system. An indoor thermostat in a key room determines the maximum temperature to be maintained.

Where a large building is to be zoned, a separate vacuum control valve is used for each zone.

R. & B. Auxiliary Clothes Dryer

Announcement of the Thermo Dryer, an auxiliary clothes-drying attachment for its small sectional



Boiler with clothes dryer

boilers, has been made by Richardson & Boynton Co., New York, N. Y. The device is coupled to the company's small sectional boilers as an integral part of the heater. Drying space is equivalent to 9 ft. of clothes line, and about 2 hr. are required for drying. The outlet temperature for normal operation is about 130°, and a damper is provided to reduce heat loss when not in operation. The boilers range in capacity from 200 to 625 sq. ft. of direct radiation.

Inverted Steam Trap

A self-venting, inverted-bucket steam trap with a large water capacity, designed for installation on all types of kitchen and hospital equipment or other service where a large volume of condensate is to be handled, has been announced by Jas. P. Marsh & Co., Chicago, Ill.

The trap, which is simple in construction and operation, discharges intermittently. Between cycles of operation the float is in the raised position, being filled with steam and air, and the valve pin is closed against the seat. After condensate rises in the trap body, the weight of the water forces the steam and air out through the vent opening in the top of the float, causing the latter to drop. When the condensate has drained from the trap, it fills with steam, causing the float to rise once more and the valve pin to close against the seat.

All piping connections are made to the body, which permits removal of the trap cover without disturbing piping. The operating mechanism lifts out with the trap cover, making the parts accessible.

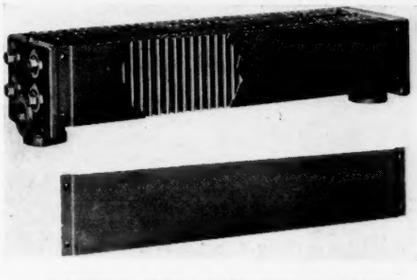
The trap, which is made in ½ in., ¾ in. and 1 in. sizes, is suitable for working pressures up to 125 lb. per sq. in.

Murray Radiator

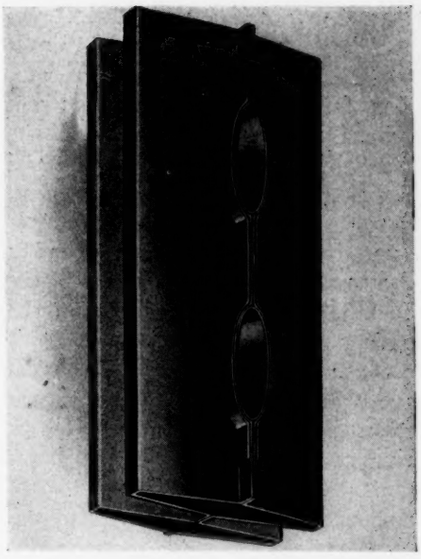
The American Radiator Co., 40 West 40th St., New York, has announced several improvements in the company's line of Murray radiators.

The new models of the radiators are designed with open fins to prevent dirt accumulation and to expedite cleaning. There are three flues per inch of length, and this wider spacing, combined with the use of oval tubes, tends to create a desirable self-cleaning effect. It is also claimed that the heat output per inch of length is higher than was the case with the previous models.

The company has published a



Cutaway view of Murray radiator showing flue construction



Close-up of flue construction of Murray radiator showing how flues are connected to oval steam chambers with welded clips

catalog containing output ratings of the complete line calculated from tests at the company's laboratory.

Manometer for Checking Flow Meters

The Meriam Co., Cleveland, Ohio, has placed on the market a high range manometer, Model 391. The instrument is designed for checking the accuracy of recording flow meters.

The device has a steel frame, stainless steel gland packers, plate glass cover, and is tested to 1000 lb. hydraulic pressure. Mercury, water, or Meriam No. 3 fluid, having a specific gravity of 2.97, may be used. With the No. 3 fluid the manometer will read 100 in. of water. The standard instrument is 42 in. high, 6 in. deep, 4½ in. wide and weighs approximately 20 lb. Special instruments of any length and for higher pressures can be supplied.

Penn JSI Stack Safety and Ignition Control

The Penn Electric Switch Co., Des Moines, Ia., has announced the Type JSI stack safety and ignition control, designed as a primary control for oil burners. Without changes in the instrument it can be

used for continuous ignition, performing safety requirements to protect against danger in case of ignition failure, flame failure, current failure, or failure of any part of the control itself. The ignition control functions include completing the ignition circuit when the motor is started and ignition cut-off after combustion has taken place. It may be mounted in the stack or on the burner draft tube.

Since it is designed for line voltage operation, the device requires a room thermostat and limit switch to give complete control of all functions. It is rated to carry ½ hp. motors. The unit is similar to the Type JS stack and lock safety switch, and uses the same construction which has been used in the control of continuous ignition burners.

Dunham Average Temperature Control

A new type of control for the company's differential vacuum heating system has been developed by C. A. Dunham Co., Chicago, Ill., and designated the "average temperature control." Prior to this time steam supply was governed by a thermostat located in a key room, or by a few such thermostats in strategic positions. When more than one thermostat was used, the one registering the lowest temperature at any time would govern the operation of the steam supply valve. Thus, the key room control actually was controlled from one point.

The new control consists of four general types of equipment, apart from the regular equipment of the differential system: (1) Nine or more thermostats of the potentiometer type located throughout the building or zone and connected to a proportioning motor valve so that the steam supply varies in accordance with the average heat demand of the building; (2) a governing panel with indicator, which can be set either for automatic or manual control; (3) an electrical controller which starts and stops the differential vacuum pump so that the pressure in the return is always lower within fixed limits than the pressure in the supply mains, the pressure differential varying with the particular building in which system is installed; (4) a proportioning motor valve. At any position the motor determines the maximum possible opening of the valve. The minimum opening may be determined by the difference in pressure between the supply and return.

Mineral Coated Welding Electrodes

A line of arc-welding electrodes, having a heavy all-mineral flux coating, has been placed on the market by the Metal & Thermit Corporation, acting as sales agents for the American Murex Corporation, the manufacturers. Both companies have offices at 120 Broadway, New York, N. Y.

The new electrodes, known as Murex, are made 18 in. long and in types and sizes suitable for a wide range of horizontal, vertical, and overhead welding operations.

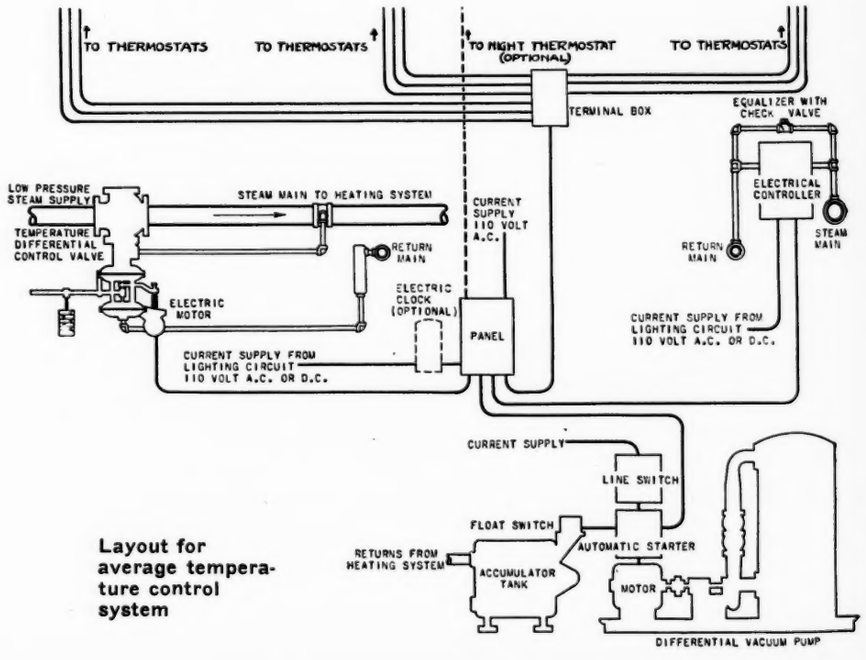
A feature of the electrode is a patented spiral winding of asbestos yarn which serves to hold the heavy mineral flux to the core so that it is not injured if the electrode is bent. The manufacturers state that the all-mineral ingredients in the flux on the electrode cause it to burn without objectionable fumes or smoke, while the thickness of the coating enables the operator to ride the electrode on the work, thus letting the flux coating space the arc.

Crane Manufactures Boilers and Radiators

Crane Co., 836 S. Michigan Ave., Chicago, Ill., is manufacturing a line of cast iron heating boilers and radiators. Production of this equipment marks the entrance of Crane Company into this field.

Boilers are of the square sectional type for commercial and residential use, and of the round type for residential use. Both styles are completely covered with jacketing, finished in two-tone green, and insulated with asbestos cell. Either coal or oil fuel may be used.

Radiators are available in the usual sizes and types. The tubes are hypocycloidal in section, resembling a square, the sides of which have been pushed into a sharply concave shape with rounded corners. This increases the flue area between the tubes and provides sufficient internal tube area for the heating medium. A shield to deflect warm-air currents away from the wall is obtainable.



Layout for average temperature control system

With the Manufacturers

Dow Chemical Company Publishes Valuable Booklet on Dehumidification

The Dow Chemical Co., Midland, Mich., has published Special Development Bulletin No. 16, entitled *Dehumidification of Air with Calcium Chloride*. The bulletin consists of 40 pages outlining the properties of calcium chloride, rate of water absorption, calculations, summary of problems, and discussion and conclusion.

The book, which is available on request, is probably as valuable a piece of work as any recently published in the general field of air conditioning. It is entirely a piece of engineering literature, and the extremely small amount of sales talk is of the informative type rather than generalizations. There are 25 graphs and flow diagrams, and numerous tables.

J. M. Robb, for the past 25 years connected with the Herman Nelson Corporation, is now operating his own sales agency, Robb Inc., 712 Peoria Life Bldg., Peoria, Ill., handling Robbins & Myers conversion burners and Electrolux refrigerators.

Silent Glow Oil Burner Corp., Hartford, Conn., has opened a new factory branch at 790 Commonwealth Ave., Boston, Mass. F. W. Chamberlin will head the branch, and J. E. Gould is supervising engineer. The new branch has a showroom 75 ft. by 40 ft.

General Heat & Power Corp., New York, has removed its offices to Room 1105, 101 Park Avenue from 730 Fifth Avenue.

Bon-Air Radiator Corp., Boston, Mass., has appointed Arthur J. Holmes, of New Haven, Conn., sales manager. Prior to his new appointment, Mr. Holmes had completed his eighteenth year with the Richardson & Boynton Company. Mr. Holmes will make his headquarters in Boston.

Kleenaire Filter Co., Stevens Point, Wis., announces the appointment of R. C. Chelius as eastern representative, with headquarters at 31 Palmer Ave., Kenmore, N. Y.

Hoffman Specialty Co., Inc., Waterbury, Conn., announces that Guy Hutchinson has rejoined its staff as general sales manager. Charles V. Haynes will continue as vice-president, in charge of the engineering and sales promotion of new products.

Carrier Corp., Newark, N. J., and *Worthington Pump and Machinery Corp.,* Harrison, N. J., announce that a reciprocal working arrangement has been effected by Worthington Pump and Machinery Corporation and Brunswick-Kroeschell Company, division of Carrier Corporation, under which the combined facilities and services of both organizations are made available to users of refrigerating and air conditioning equipment.

L. W. Grothaus has been appointed general representative of the Allis-Chalmers Manufacturing Co., Milwaukee, Wis., succeeding C. E. Searle. Mr. Grothaus has been affiliated with the Allis-Chalmers Manufacturing Company for the past twenty-eight years.

The Bryant Heater & Mfg. Co., Cleveland, Ohio, held its annual sales conferences during the weeks of January 11 and January 18. Instead of the entire sales organization meeting in Cleveland as has been the custom in the past, regional meetings were held in Philadelphia, Cleveland and Chicago.

Davis A. Fleming & Co., Inc., heating and power contractors, has moved its offices and shop to larger quarters at 2013 Sansom St., Philadelphia, Pa.

J. F. Mehlhope, formerly Chicago district sales manager for Central Alloy Steel Corporation, and later for Newton Steel Company, has joined the sales force of the Chicago Steel Service Company, warehouse distributors for Republic Steel Corporation.

Blaw-Knox Co., Pittsburgh, Pa., announces the appointment of M. I. Dorfan as manager of its dust collector division. The Blaw-Knox Company is entering the dust collector field with two lines of equipment.

John S. Webb, formerly associated with Alfred Kellogg, consulting engineer, has been appointed general sales manager of Webster Tallmadge & Co., Inc., New York.

Champion Range Oil Burner Co., 322 Chelsea St., Everett, Mass., has completed negotiations with the Champion Products Co., Worcester, Mass., to distribute the Champion Range oil burners within a radius of twenty-five miles of Boston.

Filtrators Co., Inc., Chicago, Ill., has appointed Harlan W. Bird, 111 W. Washington St., Chicago, sales agent in Northern Illinois, Indiana, Wisconsin and Eastern Iowa.

James P. Ferguson has been appointed assistant general manager of sales of Reading-Pratt & Cady Co., Bridgeport, Conn., an associate company of the American Chain Company, Inc. Mr. Ferguson has been connected with the Reading-Pratt & Cady Company for the past ten years in the capacity of sales engineer.

Crane Co., Chicago, Ill., at its annual meeting elected E. M. Ashcraft, Jr., and E. A. Russell directors of its company. The retiring directors and officers of Crane Company were reelected.

Harry H. Brown has been elected assistant vice-president of the Heating and Plumbing Finance Corporation, a division of American Radiator & Standard Sanitary Corp., New York. Mr. Brown was formerly manager of this corporation's western office in Chicago.

Ames Iron Works, Oswego, N. Y., a division of Pierce, Butler & Pierce Mfg. Corp., New York, has appointed Louis Barnes general sales manager.

Schutte and Koerting Co., Philadelphia, Pa., has issued Bulletin 16-G, describing steam and air jet fuel oil burners, manufactured by the company. The bulletin contains connection sizes, capacities of burners, and other data. Standard size, eight pages.

Statistical Service

THE chart shown at the bottom of the page shows the trend in new orders of steel heating boilers (as distinguished from power and marine boilers) from 1927-1931, inclusive. The data from which this curve was plotted were furnished through the courtesy of the Department of Commerce.

Possibly the most interesting point brought out by the graph is the change in ratio between the number of boilers and the amount of square feet of heating surface (it is important to note that the square feet of heating surface is that in the boiler and not the capacity of the boiler in square feet of radiation). The chart indicates that the size of the boilers has been steadily decreasing during the period covered. The following table, which covers the period from 1919-1931, shows this even more clearly:

Square Feet of Heating Surface of Average Steel Boiler

1919 - 1931

1919	555	1926	505
1920	673	1927	479
1921	631	1928	485
1922	600	1929	458
1923	560	1930	440
1924	524	1931	442
1925	527		

The table shows that there has been a drop of 34.4% in the size of the average boiler in the last 12 years. This trend might be explained in several ways, the most obvious being that the steel heating boiler is becoming more and more of a factor in the residential market, displacing to some extent the cast iron type of heater. Another explanation is that it is quite possible that during the period of the business depression, contractors are figuring their boilers much less liberally than previously was the case.

The charts shown on page 92, also based on data supplied by the Department of Commerce, reflect the usual seasonal drop off in heating apparatus during the month of December. The only group to show an increase are the unit ventilators, which showed a rise of 2.3%, as compared with the preceding month.

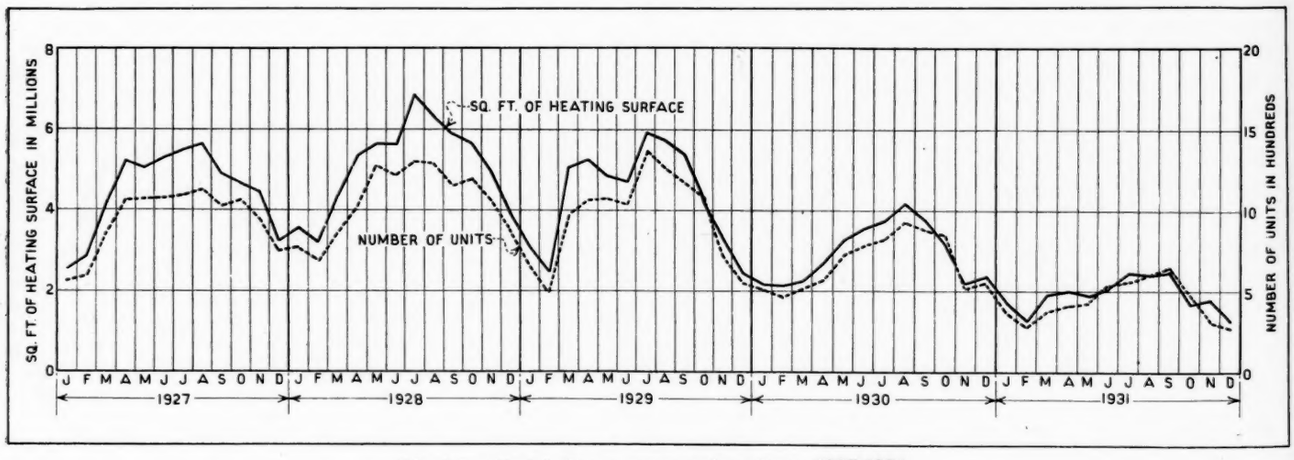
The National Association of Building Owners and Managers has compiled an estimate of the rentable area of office buildings in the United States and Canada. The compilation shows that there are 380,511,000 sq. ft. of space available in these countries, 357,964,000 sq. ft. being in the United States.

The figures for some of the large cities in this country are as follows:

New York	49,539,730
Chicago	25,469,633
Philadelphia	13,453,381
Detroit	9,337,711
Los Angeles	9,278,118
Cleveland	6,870,703
St. Louis	3,358,953
Baltimore	1,972,682
Boston	6,618,731
Pittsburgh	4,707,614
San Francisco	6,344,627

An indication of the vacancies in apartments is given in surveys covering Manhattan, made by William F. Deegan, tenement house commissioner of the city of New York. According to Mr. Deegan's report, vacancies in apartments in Manhattan have increased from 7.41% in 1927 to 14.93% in 1932.

The survey shows that the percentage of vacancies in tenement houses built under the new laws is considerably less in proportion than that in the old law tenements. The vacancies in the old law type of building are 18% of the total number of those buildings, while in new law tenements the ratio is 10.8%.


SQ. FT. OF HEATING SURFACE IN MILLIONS
NUMBER OF UNITS IN HUNDREDS
SQ. FT. OF HEATING SURFACE
NUMBER OF UNITS
0
2
4
6
8
0
5
10
15
20
J F M A M J J A S O N D J F M A M J J A S O N D J F M A M J J A S O N D J F M A M J J A S O N D J F M A M J J A S O N D
1927
1928
1929
1930
1931

Steel heating boilers—new orders from 1927-1931

(Continued on page 92)

"We have every confidence in Powers Temperature Control and would certainly install it in future mills which we build, it being the best regulation of its kind on the market."

First-Cost of Powers Control Is Often Higher

4 reasons why it costs less in the end

◀1▶ It often gives 15 to 20 years of Accurate and Dependable control without repairs of any kind.

◀2▶ Greater Steam Economy due to the fact that thermostats do not get out of adjustment and permit overheating.

◀3▶ Graduated Control, because of its greater accuracy, provides maximum comforts for occupants of rooms.

◀4▶ A minimum of service calls and the trouble and annoyance connected with them.

THE POWERS REGULATOR COMPANY

40 years of specialization in automatic heat control

2718 Greenview Ave., CHICAGO
231 E. 46th Street, NEW YORK

Offices in 37 other cities

TORONTO—106 Lombard St.

CABLE ADDRESS "THREADERS, PATERSON"

The Linen Thread Company
Mechanical Department
GRAND ST. *Paterson, N.J.*

Powers Regulator Co.,
2720 Greenview Ave.,
Chicago, Ill.

Gentlemen:-

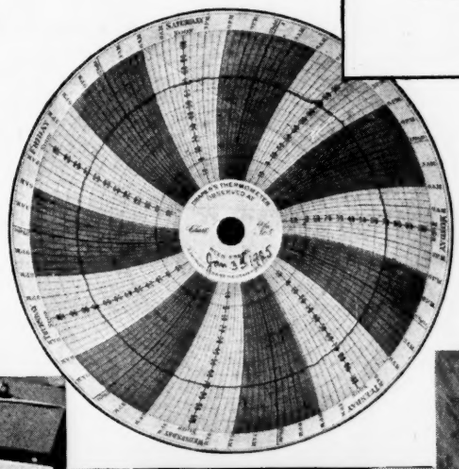
Some two years ago we investigated the different systems of heat control with a view to installing such systems in two of our mills. Would say that since that time installations have been made in the Finishing Department and Office at our Grand St. mill and in the entire new Jute Mill at Kearny, N.J.

We have kept accurate temperature records of heat in these rooms at various times and would say that the results accomplished by your control are certainly remarkable, particularly in our Jute Mill at Kearny where we need to control the temperature within very close limits on account of certain manufacturing processes. We do not believe we could have run this mill at all satisfactorily unless we had a control such as yours.

We have every confidence in Powers Temperature Control and would certainly install it in future mills which we build, it being the best regulation of its kind on the market.

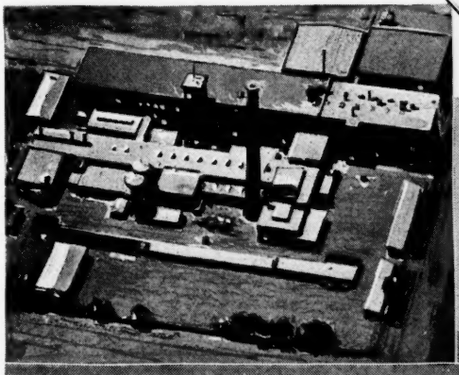
Very truly yours,
THE LINEN THREAD COMPANY

PS - WVH

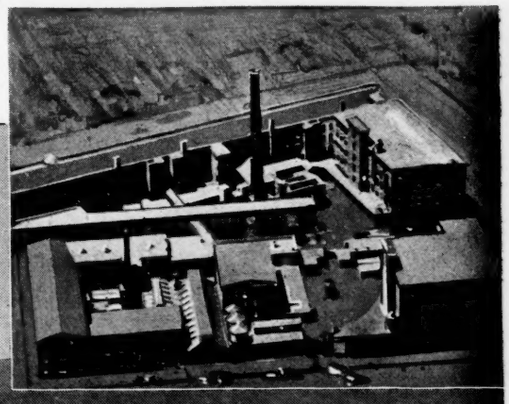


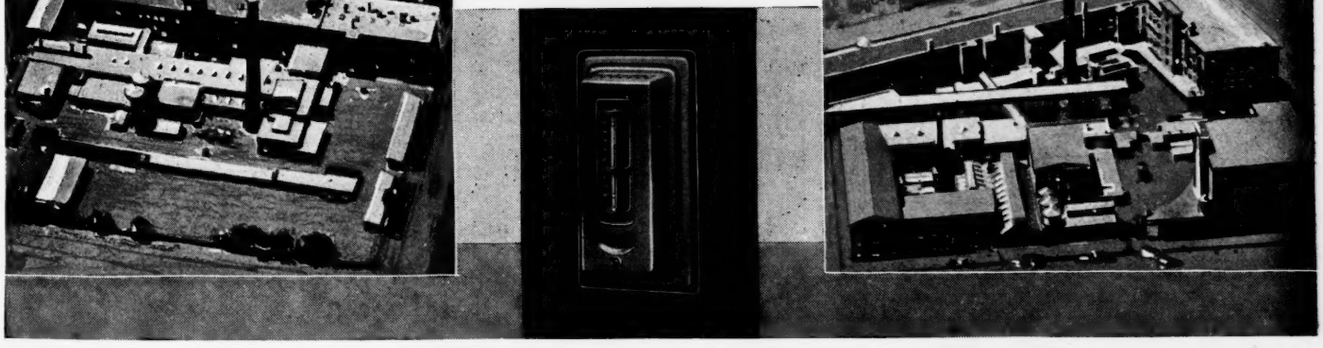
Temperature Chart shows the accurate control obtained with POWERS Regulation at the Linen Thread Company

Jute Mill of the Linen Thread Co., at Kearny, N. J.



Grand Street Mill of the Linen Thead Co. at Paterson, N. J.

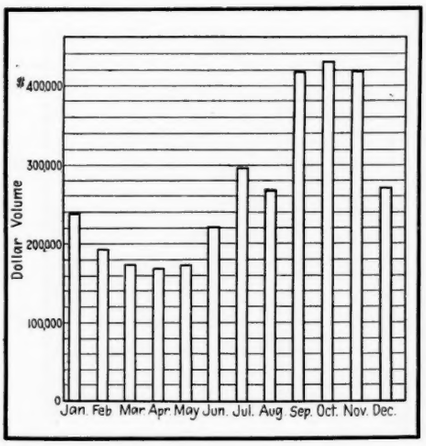




POWERS

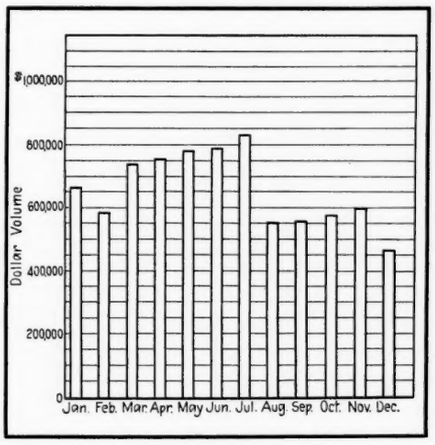
Statistical Service

(Continued from page 90)



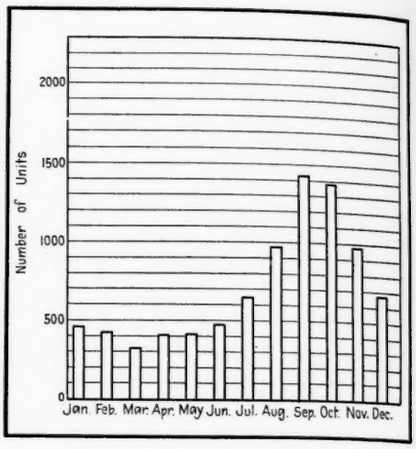
Unit Heaters (Industrial)
Orders in Dollars

December, 1931, 271,044; November, 1931, 418,712; decrease, 35.3%.



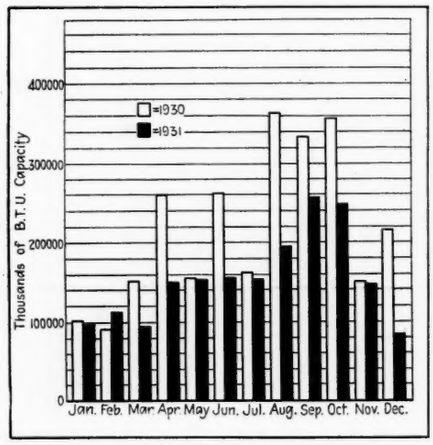
Fans and Blowers
Orders in Dollars

December, 1931, 469,513; November, 1931, 598,523; decrease, 21.6%.



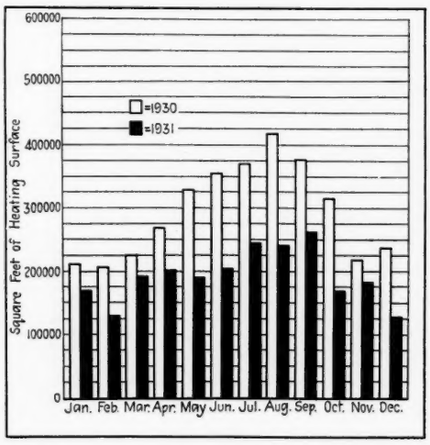
Mechanical Heating Stokers
In Number of Units Installed

December, 1931, 676; November, 1931, 976; decrease, 30.7%.



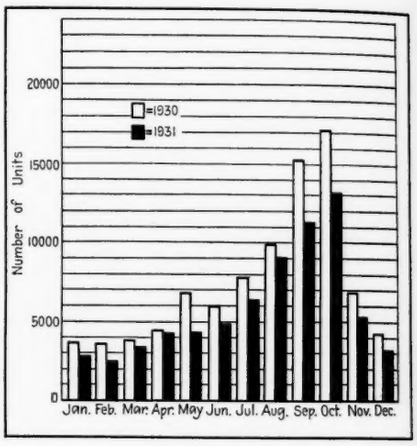
Shipments of Gas-Fired Boilers
In Thousands of B.t.u. Capacity

December, 1931, 83,230; December, 1930, 221,775; decrease over 1930, 62.4%; total, year 1931, 1,856,996; total, year 1930, 2,607,863; decrease over 1930, 28.8%.



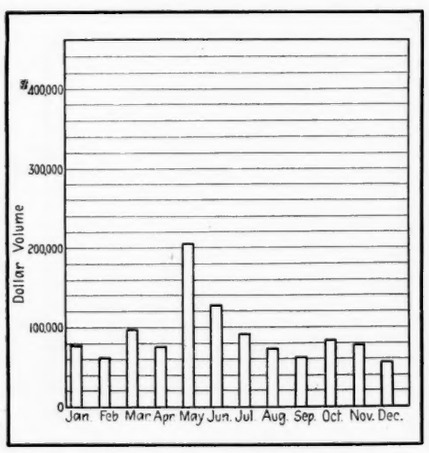
New Orders for Steel Heating Boilers
In Square Feet of Heating Surface

December, 1931, 127,285; December, 1930, 237,175; decrease over 1930, 46.3%; total, year 1931, 2,300,386; total, year 1930, 3,546,144; decrease over 1930, 35.1%.



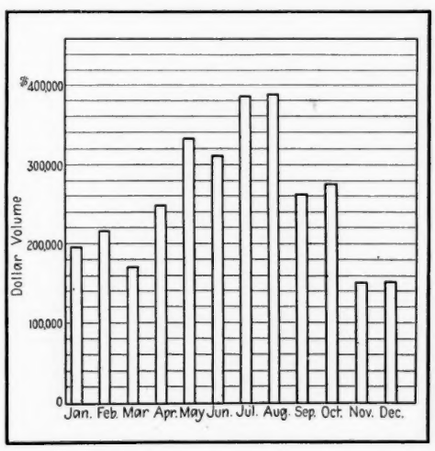
Shipments of Oil Burners
In Number of Units

December, 1931, 3186; December, 1930, 4120; decrease over 1930, 22.7%; total, year 1931, 70,863; total, year 1930, 89,022; decrease over 1930, 20.4%.



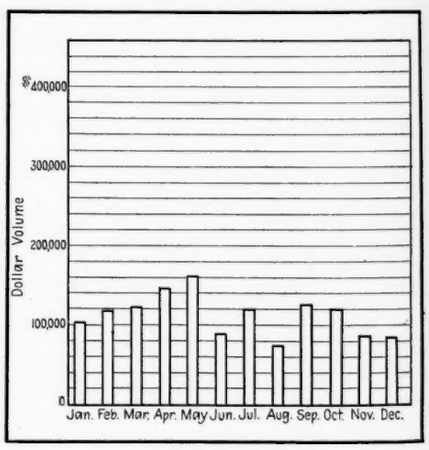
Air Washers and Air Conditioning Units
Orders in Dollars

December, 1931, 56,974; November, 1931, 78,796; decrease, 27.7%.



Unit Heaters (Schoolroom Type)
Orders in Dollars

December, 1931, 151,938; November, 1931, 151,745; increase, 2.3%.



Indirect Heating Service
Orders in Dollars

December, 1931, 85,523; December, 1930, 83,366; decrease, 2.5%.

The National Newark Building
Newark, New Jersey

Architect: John H. & Wilson C. Ely, Newark, N. J. *Htg. & Vent. Engr.:* Meyer, Strong & Jones, Inc., New York City. *General Contractor:* Starrett Brothers & Eken, Inc., New York City. *Heating Contractor:* Baker, Smith & Co., New York City. *Plumbing Contractor:* Jaehnig & Peoples, Inc., Newark, N. J.

The MAGNIFICENT National Newark Building

"Every commercial convenience and luxury that modern architecture can provide" is incorporated in the new National Newark Building. Fronting on three streets at the heart of Newark and rising to a height of thirty-four stories, the building assures to its tenants permanent advantages of sunlight and ventilation.

Every effort has been made to embody the best and the most practical in planning, in style, and in building construction and equipment, conforming with the ambition and pride of the growing, progressive, and successful city that Newark is.

The makers of NATIONAL Pipe record with satisfaction the name of the National Newark Building to be added to the long list of distinguished buildings in which, for the major uses of pipe, NATIONAL has been installed. Every such choice reconfirms NATIONAL as—

America's Standard Wrought Pipe

NATIONAL TUBE COMPANY, Pittsburgh, Pa.
Subsidiary of United States Steel Corporation

NATIONAL PIPE

Degree Days and Unit Fuel Consumption in Typical Cities

January, 1932

UNIT fuel consumption figures given below must be corrected for efficiency, heat content of fuel other than that listed below, for radiation emitting other than 240 B.t.u. per hr., and for radiation calculated on a basis other than zero outside to 70° inside.

Coal is assumed as having a heat content of 12,000 B.t.u. per lb., gas, 1000 B.t.u. per cu. ft., and oil, 140,000 B.t.u. per gal. To correct for other heating values, multiply the fuel consumption listed under the charts by the standard heat content just listed, and divide by the heat content of the fuel actually being used in the installation being calculated.

For radiation installed for any other outside temperature than zero, multiply the fuel consumption given under the charts by 70°, divided by the quantity (70 minus the outside temperature used).

To correct for efficiency, divide the fuel consumption by the efficiency of the heating installation.

For water radiation, multiply the fuel consumption which is given for steam radiation by 5/8.

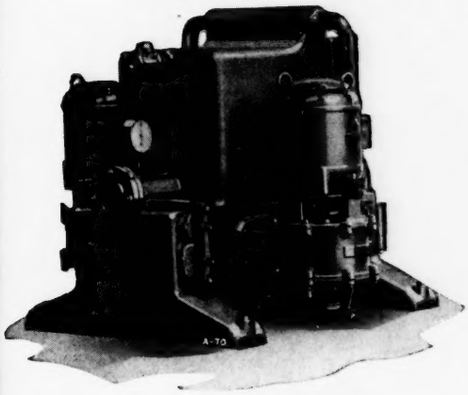
	New York	Seattle	San Francisco	Chicago	Denver	Boston	St. Louis	Pittsburgh	Minneapolis
Degree days	687	813	487	973	1156	816	781	720	1408
Total degree days since September 1	2049	2739	1766	2515	3279	2460	1904	2176	3631
Lb. of coal per sq. ft. of radiation for January	5.496	6.504	3.890	7.784	9.248	6.528	6.248	5.760	11.264
Gal. of oil per sq. ft. of radiation for January	0.471	0.557	0.334	0.667	0.793	0.560	0.536	0.494	0.965
Cu. ft. of gas per sq. ft. of radition for January	65.95	78.05	46.75	93.41	110.98	78.34	74.98	69.12	135.17

	Los Angeles	Baltimore	Philadelphia	New Orleans	Atlanta	Cleveland	Cincinnati	Detroit
Degree days	359	545	587	191	433	771	732	874
Total degree days since September 1	909	1594	1711	349	1130	2241	2036	2566
Lb. of coal per sq. ft. of radiation for January	2.872	4.360	4.696	1.528	3.464	6.168	5.856	6.992
Gal. of oil per. sq. ft. of radiation for January	0.246	0.374	0.403	0.131	0.297	0.529	0.502	0.599
Cu. ft. of gas per sq. ft. of radiation for January	34.46	52.32	56.35	18.34	41.57	74.02	70.27	83.90

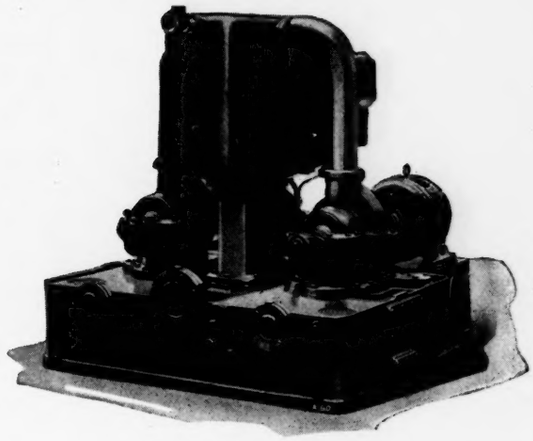
	Buffalo	Birmingham	Indianapolis	Memphis	Des Moines	Kansas City	Louisville	Galveston
Degree days	863	422	811	516	1252	1001	673	219
Total degree days since September 1	2651	956	2191*	1240	3044	2310	1773	525
Lb. of coal per sq. ft. of radiation for January	6.904	3.376	6.488	4.128	10.016	8.008	5.384	1.752
Gal. of oil per. sq. ft. of radiation for January	0.592	0.289	0.556	0.354	0.858	0.686	0.461	0.150
Cu. ft. of gas per sq. ft. of radiation for January	82.85	40.51	77.86	49.54	120.19	96.10	64.61	21.02

* *Degree Days for Indianapolis for December, 1931, were 738, and not 722, as published in the February issue.*

Fuel consumption figures must be corrected for local conditions as explained above.



Type "E" Duplex
Bulletin 600



Type "F" Duplex
Bulletin 700

Facts

Not Motor Horsepower But— Actual Power Consumption Is What Counts!

Over a Given Period, Ames Vacuum Heating Pumps Operate Less Than 40% of the Working Time.

THIS means: power is saved—less wear on working parts—and consequently, fewer repairs.

Performance records indicate, (provided the heating system is reasonably tight) that the motor on the Ames Pump operates less than 40% of the working time.

Vacuum breakdown tests, with pump shut down and inlet valve closed, show, that with an initial 26 inch vacuum, a reduction of only 3 inches occurs after a five hour period.

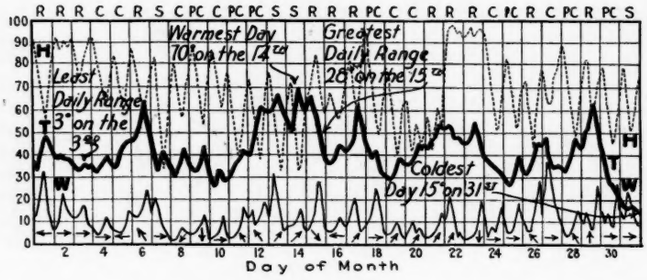
These unusual results are possible because of the distinctive design of Ames Pumps.

This is the **twelfth** advertisement in the series of twelve.

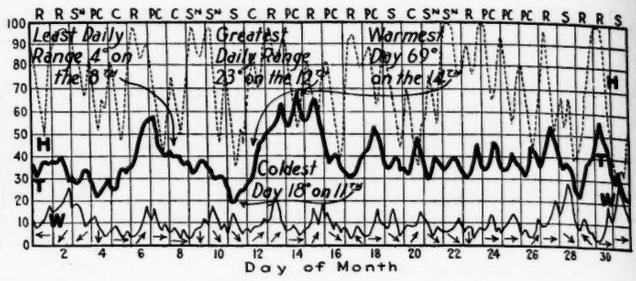
AMES PUMP COMPANY, Inc.
30 Church Street New York, N. Y.
DIVISION OF AMERICAN LOCOMOTIVE COMPANY
Representatives in All Principal Cities
Manufactured and Sold in Canada by Montreal Locomotive Works, Ltd.

The Weather

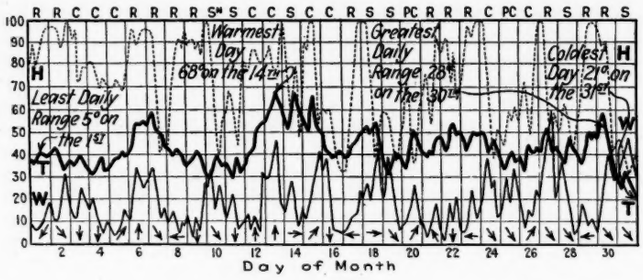
January, 1932


Warmest Day 70° on the 14th
Greatest Daily Range 26° on the 15th
Least Daily Range 3° on the 3rd
Coldest Day 15° on 31st
Day of Month

Pittsburgh

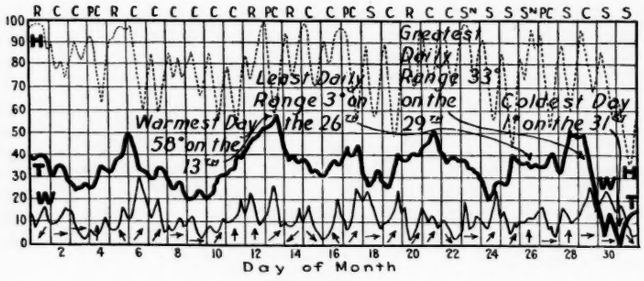

Least Daily Range 4° on the 8th
Greatest Daily Range 23° on the 12th
Warmest Day 69° on the 14th
Coldest Day 18° on 11th
Day of Month

Boston

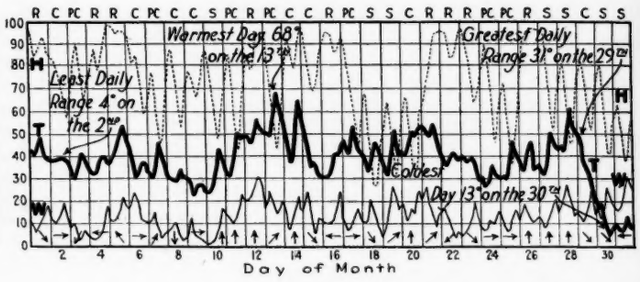

Warmest Day 68° on the 14th
Greatest Daily Range 28° on the 30th
Coldest Day 21° on the 31st
Least Daily Range 5° on the 1st
Day of Month

New York

(Hourly Observation of the Relative Humidity Plotted on this Chart)


Greatest Daily Range 33° on the 29th
Least Daily Range 3° on the 26th
Warmest Day 58° on the 13th
Coldest Day 1° on the 31st
Day of Month

Chicago


Warmest Day 68° on the 13th
Greatest Daily Range 31° on the 29th
Least Daily Range 4° on the 2nd
Coldest Day 13° on the 30th
Day of Month

St. Louis

	New York	Boston	Pittsburgh	Chicago	St. Louis
Highest temperature, deg. F.	68	69	70	58	68
Date of highest temperature	14	14	14	13	13
Lowest temperature, deg. F.	21	18	15	1	13
Date of lowest temperature	31	11	31	31	30
Greatest daily range, deg. F.	28	23	28	33	31
Date of greatest daily range	30	12	15	29	29
Least daily range, deg. F.	5	4	3	3	4
Date of least daily range	1	8	3	26	2
Mean temperature for month, deg. F.	42.8	38.8	41.8	33.6	39.8
Normal mean temperature for month, deg. F.	30.9	27.9	30.5	23.7	31.1
Total precipitation, this month, inches	4.23	4.24	4.80	2.20	2.50
Total snowfall, this month, inches	1.0	6.2	Trace	1.9	0.1
Normal precipitation, this month, inches	3.66	3.61	3.05	1.90	2.34
Total wind movement, this month, miles	12,497	6696	7501	8486	9186
Average hourly wind velocity, miles	16.8	9	10.1	11.4	12.3
Prevailing direction of wind	N.W.	W.	W.	W.	S.
Number of clear days	6	6	4	7	9
Number of partly cloudy days	4	8	7	3	5
Number of cloudy days	21	17	20	21	17
Number of days with precipitation	16	18	16	13	12
Number of days with snowfall	2	5	0	3	1
Snow on ground, at end of month	None	None	None	None	None

Plotted from records especially compiled for HEATING AND VENTILATING by the United States Weather Bureau.
Heavy lines indicate temperatures in degrees F. Light lines indicate wind in miles per hour.
Broken lines indicate humidity in percentage from readings at 8 a. m., 12 m., and 8 p. m.
S—clear, PC—partly cloudy, C—cloudy, R—rain, Sn—snow. Arrows fly with prevailing directions of wind.

Heating and Ventilating

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A. A. Berestneff

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G. W. Nigh

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W. J. King

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Filtered Air for Hay Fever and Asthma Treatment
Frank C. Taylor

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Air Conditioning Conference at Case School

Proposed Building Regulations for New York

Air Conditioning—Part XVIII Condenser Water for Refrigeration

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